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AUTOMOBILE ENGINEER

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Instruments and Accessories

THERE are welcome signs that common sense is prevailing over the stylist in the matter of facia panel layout. Further, the fact has been re-discovered that the tip of a pivoted pointer moves around a circular arc, and not around the edges of a square or along a straight scale. If there is still cause for legitimate complaint that in some cases the figures demand close scrutiny before they can be readily recognized, it may be reasonably countered by the argument that in all conditions of actual driving the message of the instrument is conveyed at a glance by the angular relation of the pointer, and not by actual reading of the scale. The form of the figures thus becomes primarily a question of good taste, in which, however, simplicity is a major virtue, as in other spheres.

Dashboard Lighting

In general, then, there has been a marked improvement in these matters. Efforts might well be directed now to the question of instrument lighting, in which there are still wide variations. The major fault is excessive brightness, particularly when edge-lighting from the side is provided. Probably the majority of drivers would prefer to drive with the instruments illuminated, but for the fact that in so many cases an unpleasant glare is the principal effect. Doubtless the effect is more pronounced to some drivers than to others, but in general there seems to be a need for substantial improvement. There are in fact some examples that are practically impossible to tolerate, and a night journey is punctuated by periodic groping for the panel light switch in order to make an occcasional check that all is well. With the present fashion for rows of identical knobs along the facia, such action is quite likely to result in switching on the fog lamp or starting the screenwipers.

It should not prove impossible to devise an acceptable lighting standard for all vehicles. Alternatively, although it is rather a refinement than a necessity, rheostatic control could be more widely embodied. In either case, however, the panel light switch should be distinctive by position or " feel," so that those who prefer to drive "blind" may make their periodic checks without a preliminary process of conscious selection that diverts attention.

On wider ground, an occasional review of instrumentation and standard accessories as a whole, is worth a little time. For a given budget on these components, more thought might well be given to the most useful selection from the viewpoint of practical service in rendering the vehicle more roadworthy. It is, of course, easier to add than to subtract, and personal opinion becomes a considerable factor when strictly non-essential items are under review.

It might be suggested, however, that the addition of a screen washer would more than offset the subtraction of a clock, indicator warning lights, ammeter and "courtesy lights." This device is, in fact, second only to the screen-wiper itself as an adjunct to safe driving. There appears to have been a marked increase in "mud slinging" in recent years, doubtless in part owing to the increased traffic density, but also apparently owing to the number of medium weight trucks in an indifferent state of maintenance, particularly with regard to rear mudguarding. These vehicles usually travel quite fast, so that longer periods of "following" occur. Overtaking such traffic on a road drying after rain can be a manoeuvre fraught with more than a mere element of danger, and in these conditions a screen washer is of particular value. Even in dry weather, visibility is appreciably reduced towards the end of a long run by the accumulated dust, and in certain lighting conditions and at night the value of a clean screen is not fully appreciated until the sharp contrast provided by a washer has been experienced. The screen washer, in fact, has a strong claim to be regarded as an essential standard fitting on every car.

The cynic would, of course, find no difficulty in the matter of subtraction. If he chose even an approximation to accuracy as a basis for elimination, the petrol gauge would be very near the head of the list. Indeed it has been there since it first appeared on the facia. Were it not a legal requirement, however, it would probably be the speedometer that would hold the first place.

Accuracy Standards

While accuracy is of no great importance with most instruments, since to the ordinary driver they are there to show tendencies and not absolute readings, there can be no reasonable excuse for the gross exaggerations common in the upper speed range of many speedometers. The performance of all popular cars is usually widely known as a result of independent testing, so that nothing is gained by indicated speeds that are manifestly absurd. The pseudo-psychological argument that many drivers cruise safely at 40 m.p.h. while happily travelling at an indicated 50

m.p.h. is also unworthy of a moment's consideration. No convincing reason has so far been advanced for the aberrations that seem to be inseparable from this instrument.

Although barely within the scope of this subject, the location of the dipswitch might well be re-considered. The foot dipper, now almost universal, seems peculiarly ill adapted to cars that require frequent gear changing. Always excepting experienced organists, three pedals are enough for two feet to control with certainty, whereas a properly placed finger tip switch that can be operated without lifting the hand from the wheel meets all needs. Many would perhaps echo the remark of an experienced driver concerning the foot dipper—"I always change gear on it."

Persian Oil

AS the journal of an industry vitally concerned with fuel supplies, *The Automobile Engineer* has from its earliest days pressed for an indigenous fuel industry. As a country entirely without oil, it was regarded in an oil age, as essential that a Home produced fuel industry should be developed concurrently with the growth of road vehicle usage. In January, 1918, the following comments appeared in *The Automobile Engineer* :—

"In pressing for a proper utilization of our coal, we put the matter forward from the viewpoint of the automobile and aircraft industries. We encourage the new methods becuase the spirit could be produced within these Islands. There is, however, a phase of the subject that is of greater imperial importance, namely, the provision of a home supply of oil fuel for the Navy, in which connection the newer process assumes tremendous magnitude.

"The way in which Official ineptitude has been tolerated in this question of fuel supply stands out in all its menacing inefficiency, etc.

"Intelligent and vigorous action is necessary because powerful interests have to be combated.

"Those supporting obsolete methods of coal treatment are running directly counter to the National interests."

Dealing again in 1918 with the Report of the Fuel Research Board, we made the following comments :—

"One extraordinary sentence occurs in the Report that may well give rise to the gravest apprehensions. It is laid down in the Report as a condition regarding any improved system, that *no new carbonization scheme can be justified economically, if it can only live by poaching on the preserves of existing industries.* Careful consideration of this amazing dictum reveals only two possible alternatives, namely either that the gate to progress is being frankly shut and locked behind the vested interests, or that the author has painfully obscured views as to the potentialities of the sentence. We feel that such a statement needs a thorough explanation and justification, and we call upon the Board to define explicitly the limitations it would impose upon new carbonization methods and so remove the many grave doubts engendered by the statement.

"The Report further states that '*the Board would regard it as a great misfortune if the establishment of a Government organization for fuel research were to result in the discouragement or limitation in any way of the activities of outside workers.*' The absence of any evidence of interest in or assistance given to these workers would appear somewhat to qualify the sincerity of these aspirations. We are afraid that the report of the Fuel Research Board leaves us very much where we were, save perhaps that progress has received a little more Official discouragement.

"The tendency, not uncommon in Government departments and its Committees, would appear to be to conserve at all costs the rights of vested interests, in this case, of course, those vested in fuel."

Although appearing as long ago as the first world war, these views were stressed and amplified many times during the ensuing years. The consistent policy of neglect would now appear to have placed us in a position that is full of highly disturbing potentialities. It is nevertheless a policy that has been consistently pursued, even in the matter of forcing steam driven vehicles off the road by short sighted regulations. Gas producers received no support and were handicapped by the same regulations. Experimental installations of Town gas stored in high pressure bottles were discontinued, and nothing was done to develop the use of methane from sewage disposal work, collieries, or coke ovens. Low temperature carbonisation of coal was allowed to drift along struggling alone against prejudice and lack of support.

A proprietary creosote based fuel was developed and commercially produced on a substantial scale. With a special dual carburettor employing petrol starting and idling, it gave satisfactory and economical operation. The sponsors activities were however continually obstructed by official action, even including the altering of the restricting order. The only conclusion must be that in the official view, Home produced fuels were not desired. During the recent war, by means of hydrogenation and other schemes, Germany managed to supply both its war and civilian needs. There is little doubt that had foresight been shown in the very early days of the industry, this country would now have been in possession of a fuel industry that would at least have taken a good deal of the sting out of the present trend of events.

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BEDFORD "S" TYPE CHASSIS

A Range of Cab-over-engine Vehicles of 7 ton Rating

THE new "S" models listed, are additions to the present range of Bedford vehicles. They are designed on the cab-over-engine principle to obtain equal tyre loadings front and rear with twin rear tyre equipment, and to provide the maximum loading space in relation to their over-all vehicle length. The drop side, platform and end tipper vehicles, will carry a 7-ton payload when fitted with Vauxhall-built bodies. With a taxation weight of less than 3 tons, this brings the vehicles into the motor class with a permitted "domestic" speed limit, under present regulations, of 30 miles per hour.

MODELS AVAILABLE

Model	Supplied as	Wheelbase
SSZ	Chassis only	9 ft. 8 in.
SSC	Chassis with cab ..	9 ft. 8 in.
SST	End-tipper truck ..	9 ft. 8 in.
SLZ	Chassis only	13 ft.
SLC	Chassis with cab ..	13 ft.
SLD	Dropside truck ..	13 ft.
SLP	Platform truck ..	13 ft.
SA	Articulated tractor unit chassis with cab ..	7 ft. 2 in.
SB	Coach chassis only ..	17 ft. 2 in.

A tractor unit chassis is designed solely for use with the Bedford-Scammell articulated vehicle, and a coach chassis has been developed as the basis of a 32-seater luxury coach or bus. This chassis is built to comply with domestic P.S.V. conditions of fitness regulations. Adaptability is one of the particular features of this range of vehicles.

SPECIFICATION

ENGINE. Six cylinder, 4927 piston displacement. Bore and stroke 3⅞ × 4¼ in. Maximum b.h.p. 110 at 3200 r.p.m. Maximum brake torque 234 lb./ft. at 1200 r.p.m. Compression ratio 6 to 1. Overhead valves push rod operated. Seven main bearings.

CLUTCH. Borg and Beck 12 in. diameter single dry plate.

GEARBOX. Four forward speeds and reverse. Top, third and second ratios are synchromesh. Ratios, 7.06 to 1, 3.332 to 1, 1.711 to 1 direct, 7.06 to 1 reverse.

REAR AXLE. Fully floating with straddle mounted pinion. Alternative ratios 5.833 to 1 and 6.8 to 1.

SUSPENSION. 2½ in. wide semi-elliptic laminated variable rate springs.

STEERING. Semi irreversible worm and sector. Solid spoked steering wheel, moulded rubber covered rim.

BRAKES. Hydraulic operation front, mechanical rear. Vacuum servo assisted.

DIMENSIONS. Wheelbase, Model SS, 9 ft. 8 in.; SL, 13 ft. 0 in.; SA, 7 ft. 2 in. Track, front 6 ft. 0¾ in., rear 5 ft 7⅛ in. all models.

Engine

Made of Chromidium alloy iron the cylinder block is cast integrally with the crankcase. The sump is attached to the lower face of the crankcase at the same level as the bottom of the main bearing caps. Webs stiffen the structure and provide rigid support for the main bearings. The cylinder bores are fitted with full length slip-fit centrifugally cast alloy iron liners. These are located by a flange formed on the top of each liner seating in a recess machined in the top face of the block, and they are held in position by the cylinder head gasket. The clearance between liner and bore is 0·0025 in. to 0·0005 in.

The crankshaft is forged from high tensile steel with the counter-weights integral with the webs, and the journals and crankpins are induction hardened. A spring-loaded lip-type oil seal fitted in the timing cover is provided at the front end of the shaft, and the rear end is machined to provide a screw-type oil retainer. Each of the seven main bearings is fitted with

Short wheelbase Bedford "S" type chassis.

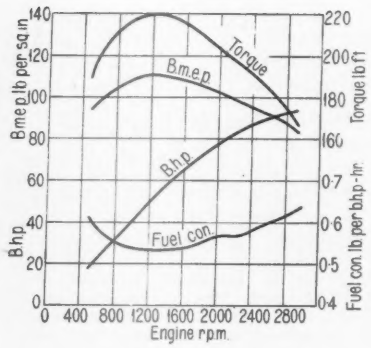
copper-lead lined, pre-finished half bearing shells. Circumferential and endwise location of each half bearing shell is by a projection on the liner which enters a slot provided in the case and end caps. End thrust is taken by white metal faced steel washers split for assembly on to each side of the centre main bearing. Circumferential location of the washers is by means of a tag which fits in slots provided in the bearing cap flanges. Broached slots in the crankcase provide transverse location for the bearing caps. Longitudinal location is effected by the high tensile steel attaching bolts which are locked by a double-ended bi-hex lock plate attached to the centre of the bearing cap by a set-screw and spring washer.

CRANKSHAFT DATA	
Mean journal diameter ..	2.7495 in.
Mean crankpin diameter ..	2.3745 in.
Mean length front bearings	1.609 in.
Mean length centre bearing	1·630 in.
Mean length rear bearing ..	1·672 in.
Mean length intermediate bearings	1·220 in.
Main bearing clearance ..	0·002–0·0045 in.
Crankshaft end float ..	0·002–0·012 in.

A single assembly is formed by the crankshaft pulley and harmonic damper which are fitted on the front end of the crankshaft, and retained in position by the combined starting dog and crankshaft nut. The harmonic damper consists of a circular weight mounted on the front face of the crankshaft pulley flange by means of a retaining ring and six shouldered rivets carried in integral bushes in the rubber inserts. Torsional vibration of the crankshaft is reduced by tuning the damper to 120 : 190 cycles per second thus giving long life and quiet running. The high tensile alloy cast-iron flywheel incorporates a separate ring gear of 120 teeth, and is bolted to the flange formed on the rear end of the crankshaft.

Oval ground skirts on the aluminium alloy pistons allow for uneven expansion, and tin plating facilitates running in. Endwise location of the full floating hollow 1 in. diameter gudgeon pins is provided by circlips fitted into the bores. The gudgeon pins are case-hardened, ground and lapped. A hard push fit in the piston is obtained at 68 deg. F. by selection.

All the rings are of the high pressure type individually cast to form. The top ring is a taper faced compression ring chromium plated on its outer face, and the second ring is a cornered compression ring. Two slotted-type oil scraper rings situated one above and the other below the gudgeon pin are fitted. With the exception of the top compression ring they are all phosphate finished to give



Power curves.

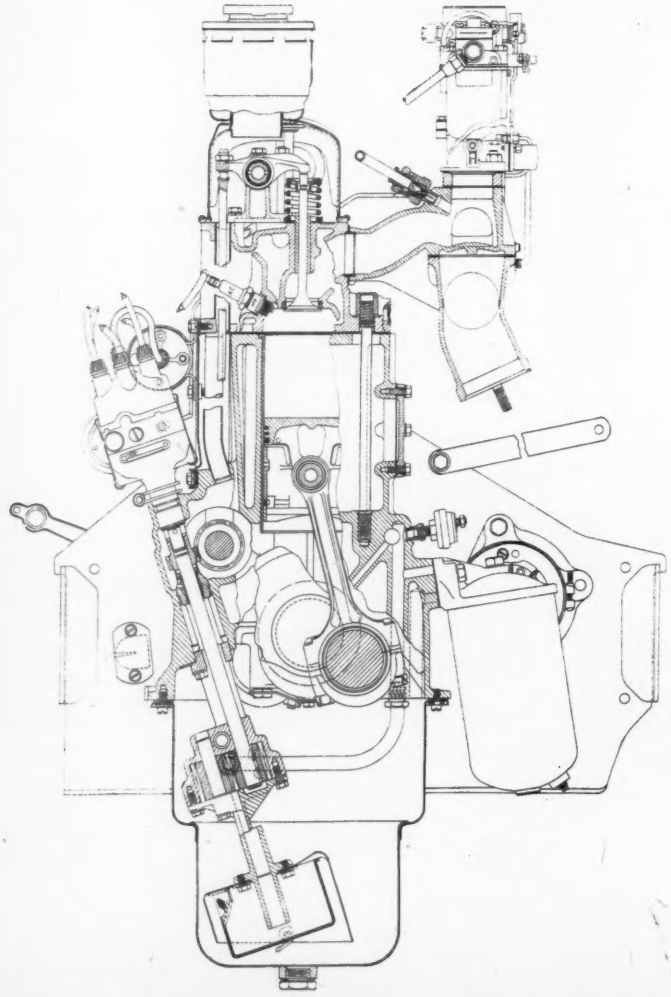
an oil retaining surface, and to improve running in qualities.

The connecting rods are H-section heat-treated steel stampings with bronze bushes in the small end for the fully floating gudgeon pins. Lengthwise drillings in the rod carry oil from the big to the small end. Faced at right angles to the axis of the rod, the big ends are fitted with steel shell copper-lead lined, pre-finished liners.

CONNECTING ROD DATA	
Length of centres	7·625 in.
Big end width	1·616 in.
Small end width	1·252 in.
Big end bearing liner length	1·402 in.
Big end bearing clearance ..	0·002–0·004 in.
Big end bearing end float ..	0·006–0·012 in.
Small end bearing clearance	0·0002–0·0011 in.

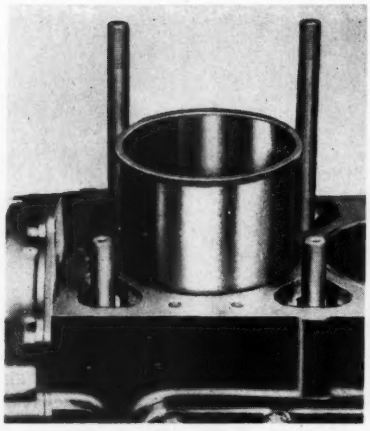
The oil sump is shallow at the rear to provide clearance for the axle. At the front end it is deepened by a second pressing which is fitted into the top pressing and welded to it. A drain plug is provided on the bottom face of the sump.

A helical-gear-type oil pump in tandem with the ignition distributor is driven at half engine speed from the camshaft spiral gear. The oil pump is protected by a wire gauze strainer in a suction cup, and has a by-pass



Engine cross section.

arrangement in case the strainer becomes clogged. A non-adjustable pressure relief valve is fitted in the pump carrier. Filtration of the oil is effected by an A.C. by-pass type filter. Oil under pressure is piped at 35 to 50 lb. per sq. in. to a main gallery drilled lengthwise through the crankcase, whence it passes through drilled passages to each main bearing. From there it passes through grooves and drilled holes to the camshaft bearings, and also through drilled holes to the crankpins, and thence to the small ends. Oil jets from holes in the big ends lubricate cylinder liners and gudgeon pins. The rocker and valve mechanism is positively lubricated by oil fed from the front intermediate camshaft bearing to a connection between the front and centre hollow rocker shafts. A connector, which incorporates an oil over-flow pipe, joins the centre and rear rocker shafts. This over-flow pipe is looped up higher than the shaft in order to maintain a constant head of oil.

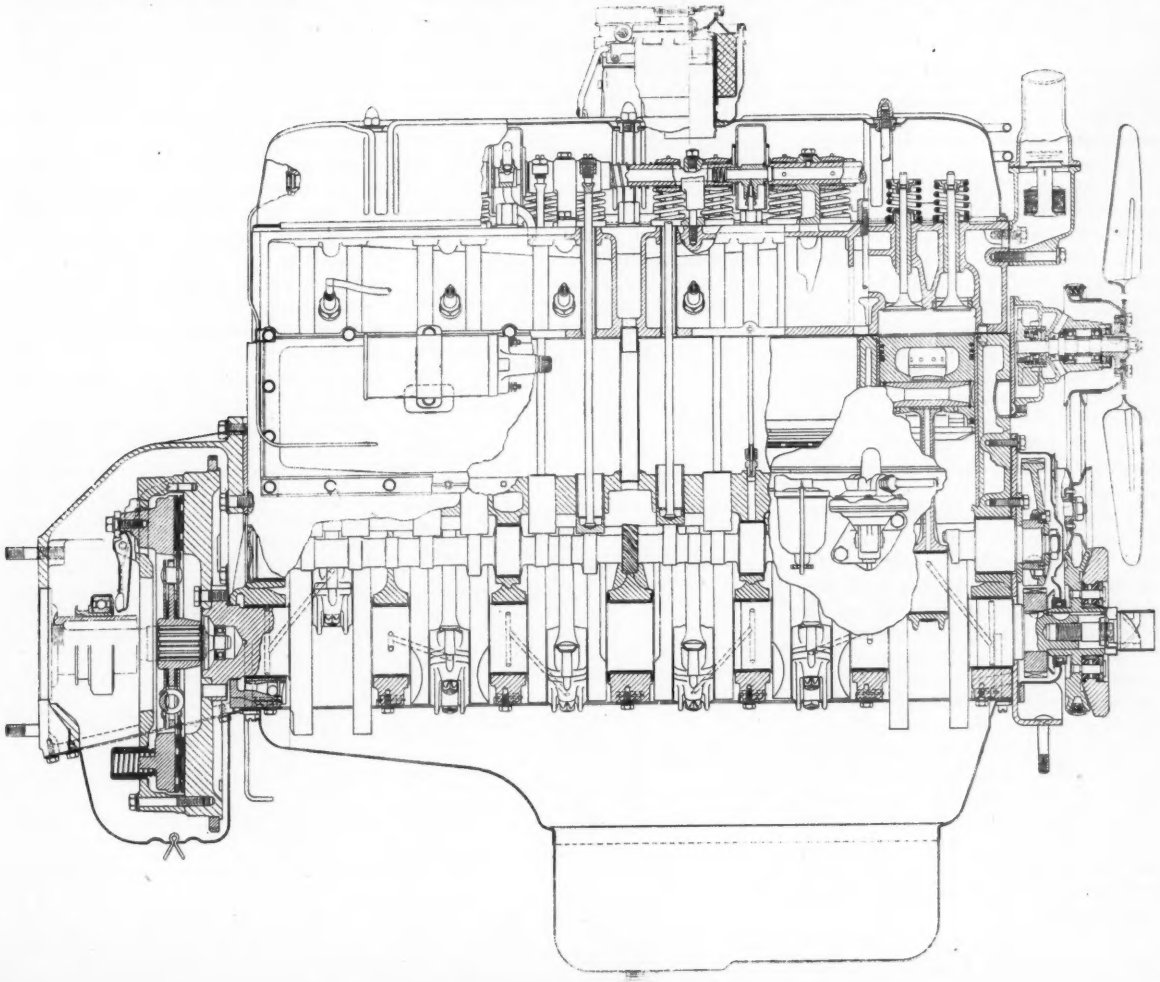


Cylinder liner.

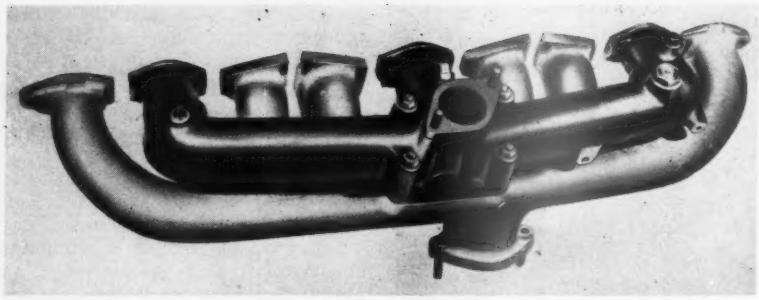
The oil filler, at the top of the rocker cover, is part of an oil bath air cleaner for the crankcase suction ventilation system which incorporates an A.C. valve at the inlet manifold. Some form of air cleaner is, of course a most essential feature for vehicles destined for dusty countries. An oil level dipstick is provided on the right-hand side of the crankcase.

All valves operate in renewable guides. The inlet stems are provided with felt seals. Valve seat inserts of Durachrome are used for the exhaust valve only. Separate exhaust passages are provided for each cylinder, and are as short as possible to minimize heat transference to the coolant. The combustion chamber is of compact design and the valves are all mounted at the same level in the top of the head. Fourteen high tensile steel studs and special lead brass washers and cap nuts secure the cylinder head to the block. Each stud is screwed into the base of the crankcase to minimize the possibility of cylinder head distortion. The joint is made good with a copper and asbestos gasket.

Chilled cast tappets with steel push rods in conjunction with bronze bushed steel rockers operate the valves. A screw and lock-nut are provided to



General arrangement of Bedford engine.



Inlet and exhaust manifold.

adjust the tappet clearance, and a single valve spring is retained on each valve by a cup and two split collars.

VALVE DATA

	Inlet	Exhaust
Material ..	50 carbon steel	Silchrome XB steel
Head diameter	1 11/16 in.	1 1/16 in.
Stem diameter	11/32 in.	11/32 in.
Lift	0·402 in.	0·402 in.
Valve clearance	0·006 in.	0.013 in.
Seat angle ..	29 deg.	44 deg.

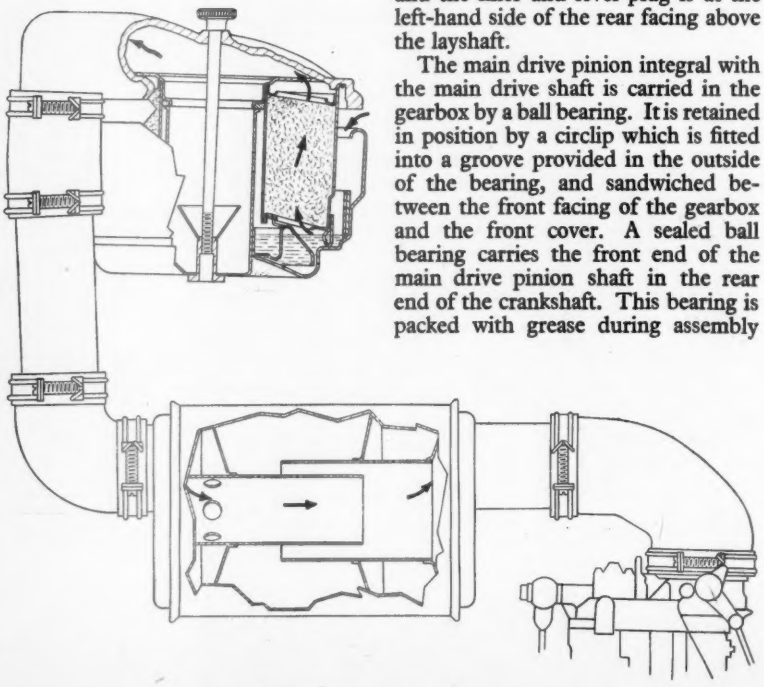
Inlet valve opens at T.D.C. Inlet valve closes at 42 deg. after B.D.C. Exhaust valve opens 32 deg. before B.D.C. Exhaust valve closes 4 deg. after T.D.C.

A heat-treated steel camshaft is supported in the crankcase by four steel-backed, white metal lined bearings. The cams are specially designed for quiet operation. A spiral gear on the camshaft drives the ignition distributor and oil pump, and an eccentric operates the fuel pump. Helical gears drive the camshaft from the front end of the crankshaft. Lubrication for the camshaft drive gears is provided by a jet of oil from the crankshaft front bearing. Timing marks are stamped on the gears.

Water circulation is from the bottom of the radiator through the pump to the front right-hand side of the cylinder block, round the cylinders, through passages at the end of the block to the cylinder head jackets, and thence through the thermostat valve to the top of the radiator. The two-piece four-bladed fan and centrifugal impeller pump are mounted on a single spindle and driven from the front end of the crankshaft at 1·3 times engine speed by an endless rubber and canvas 42 deg. V belt. Sealed ball bearing journals, requiring no lubrication carry the spindle which is fitted with a self-adjusting spring-loaded gland.

On the left-hand side of the engine the cast iron six port exhaust manifold provides a hot-spot for the three port cast aluminium inlet manifold. An A.C. mechanical pump driven by an eccentric on the camshaft delivers fuel, after it has passed through a gauze filter in an inverted glass bowl, to the Zenith 42 VIR/3 downdraught carburettor at a pressure of 2½ lb. per sq. inch.

A part throttle diaphragm type economy valve bleeds air to the jet system, and a mechanical accelerator pump is fitted. Inter-connected choke and throttle controls provide correct throttle opening for cold starting. Large inlet depressions automatically open the choke flap, as it is eccentrically hinged and spring-loaded to prevent over enrichment when opening up from cold. The engine governor works on the gas speed principle. An unusual feature of the induction system is the inclusion of a properly tuned silencer in addition to the oil bath air cleaner.



Air intake filter and silencer.

Clutch

A single dry plate 12 R.D. Borg and Beck clutch has wound yarn friction linings and a spring cushioned drive. The clutch pedal, which is carried on a bracket secured to the frame side member, actuates the forked clutch withdrawal lever through an adjustable push rod, the forked lever being pivoted on a ball-ended pin, and the thrust is transmitted to the clutch levers through a sealed ball bearing.

CLUTCH DATA

Nominal diameter	12 in.
Number of friction facings ..	2
Friction facing outside diameter ..	11·88 in.
Friction facing inside diameter ..	7·505 in.
Friction facing thickness ..	0·145 in.
Total friction area	133 sq. in.
Number of pressure springs ..	9
Assembled load	165–175 lb.

Gearbox

The gearbox is built as a unit with the engine bolted to the clutch housing in the usual manner. It gives four forward gears and one reverse gear, the top, third and second speed gears being provided with synchromesh. A centrally mounted gear shift lever gives a standard S.A.E. change.

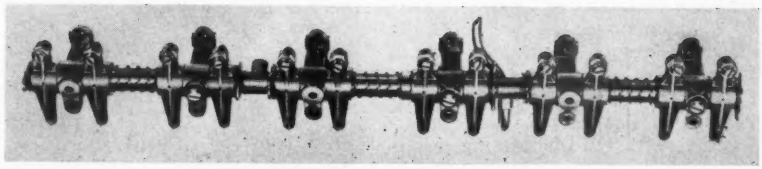
An aluminium die casting with detachable top, front, side and rear covers forms the gearbox casing. Provision is made for a standard S.A.E. six-bolt power take-off on the left-hand side, and it is sealed by an aluminium cover when a power take-off unit is not fitted. A drain plug is fitted at the bottom of the rear face, and the filler and level plug is at the left-hand side of the rear facing above the layshaft.

The main drive pinion integral with the main drive shaft is carried in the gearbox by a ball bearing. It is retained in position by a circlip which is fitted into a groove provided in the outside of the bearing, and sandwiched between the front facing of the gearbox and the front cover. A sealed ball bearing carries the front end of the main drive pinion shaft in the rear end of the crankshaft. This bearing is packed with grease during assembly

and requires no further lubrication. An extension of the main drive pinion front cover is provided over the shaft to form the support for the clutch withdrawal bearing and sleeve assembly. The inner end of the main drive pinion is integral with the outer sections of the top gear dog clutch and synchro cone. The inner section of the synchro cone is carried on the integral male section.

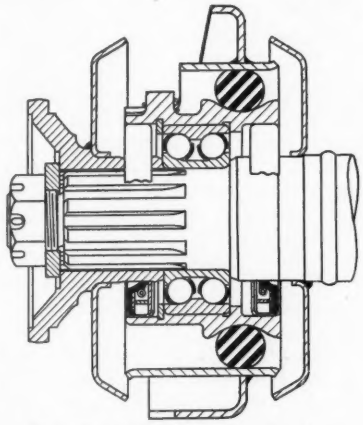
Rollers located in the bore of the main drive pinion, carry the front end of the mainshaft, and the rear end is carried by a ball bearing located in the rear wall of the gearbox. This bearing is positioned by a circlip fitted into a groove provided in the outside of the bearing, and sandwiched between the gearbox rear face and the rear cover. A clutch assembly for the fourth and third speed gears is mounted on the front splined section of the mainshaft, and retained in position by a circlip fitted into a groove in the shaft. The constant mesh, third and second speed gears are bushed with steel-backed lead bronze shells, and run on the mainshaft. They are located endwise by a shoulder on the shaft.

The third speed gear is located by the face of the fourth and third speed clutch sleeve, and the second speed gear by the face of the first speed splined sleeve. A splined sleeve supports the first speed sliding gear and is secured to the mainshaft by circular section keys. It is held on the shaft by the mainshaft ball bearing, distance washers, speedometer drive gear, and the universal flange. These are secured endwise on the shaft by a thick flat washer and a hexagon headed bolt locked by a tab washer



Valve rockers.

and screwed into the end of the mainshaft. The layshaft, made integral with the first speed gear, is carried at the front and rear of the gearbox by ball bearings. This shaft is located in the box by the front ball bearing, which is secured endwise by a circlip fitted



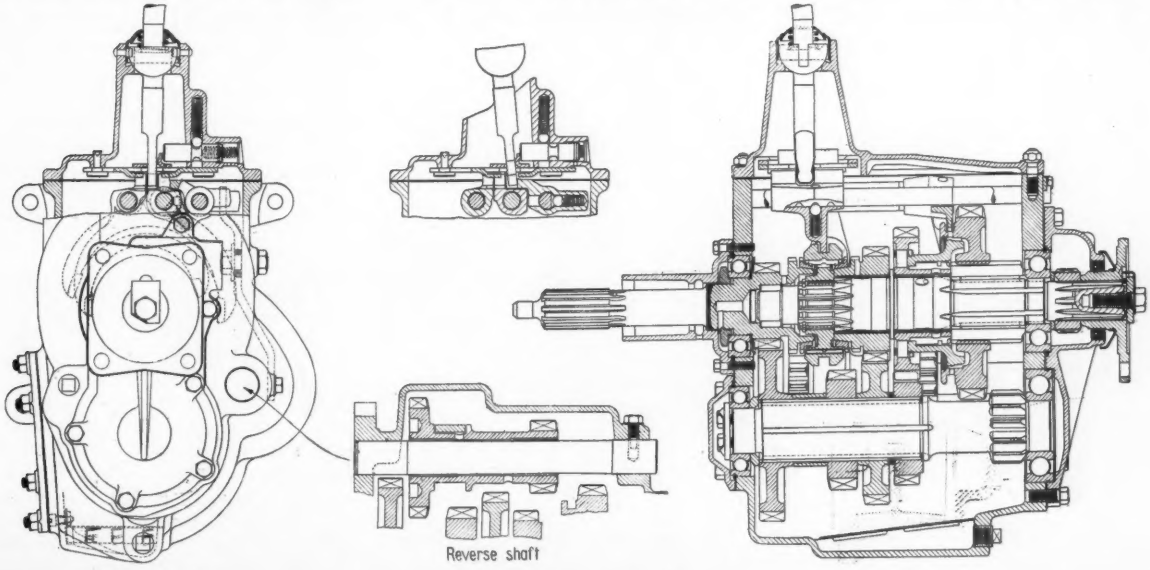
Propeller shaft intermediate bearing.

into a groove on the outside of the bearing, and sandwiched between the front face of the gearbox and layshaft front cover. Spacers separate the first reduction, reverse, third and second speed gears, which are pressed on the shaft, and driven by four long circular section steel keys. The shaft and gear assembly is secured in the front ball bearing by a thick washer and two hexagon-headed bolts secured by a lock plate and screwed into the front end of the shaft.

Steel backed lead bronze liners carry the integral reverse idler gears which run on a stationary shaft fixed into the gearbox casing. The idler gear assembly slides on the shaft for engagement of the layshaft reverse gear, and the mainshaft first speed gear. Except when in use the reverse gear assembly is stationary.

The speedometer gears are in the gearbox rear cover, which also houses both the main and layshaft bearings. Provision is made for the various sizes of speedometer driven gears required with the axle ratios and tyre sizes fitted to the vehicles. A lip type oil seal fitted in the rear of the cover works on the universal joint flange shank. This oil seal is protected from exterior dirt by a pressed steel mud slinger attached to the universal joint flange.

Three selector forks are carried by the selector rods located in the gearbox above the mainshaft gears. These forks slide on the rods which are secured and located in the box by a key plate attached to the gearbox



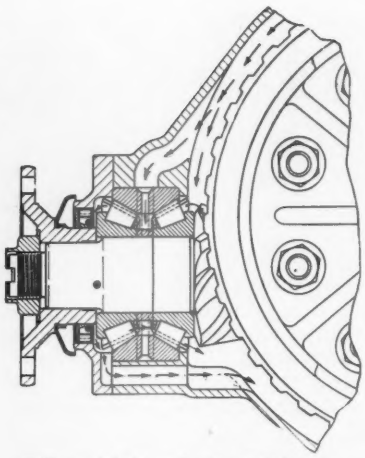
General arrangement of gearbox.

rear face by setscrews. Spring loaded balls mounted in the selector forks engaging grooves provided on the underside of the selector rods retain the gears in their operative and neutral positions. The change speed lever, together with the interlocking mechanism which prevents the engagement of more than one gear at a time, is mounted in the detachable top cover. A spring loaded ball engaging in a groove, in conjunction with a spring, helps to provide a very definite resistance to accidental engagement of reverse.

Propeller shafts

The single open propeller shaft fitted on the "SS" and "SA" models is connected to the front universal joint by a sliding coupling, each end of the shaft being fitted with Hardy Spicer needle roller joints. In the case of model "SL" two open propeller shafts are arranged in line. The rear shaft is provided with a universal joint at each end, and the rear end of the front shaft is carried in an intermediate bearing (see diagram), and the arrangement of the front joint is the same as in models "SS" and "SA", except that there is no sliding joint. On the longer bus chassis "SB", three open propeller shafts are fitted in line, with four Hardy Spicer joints. The rear ends of both the front and the intermediate propeller shafts are carried in intermediate bearings attached to chassis frame cross members. Universal joints at the intermediate positions are located just behind the bearings, and the universal joint at the rear intermediate bearing is joined to the rear propeller joint by a sliding coupling.

A thick plain washer and a slotted nut locked by a split pin secures the



Differential pinion bearing lubrication.

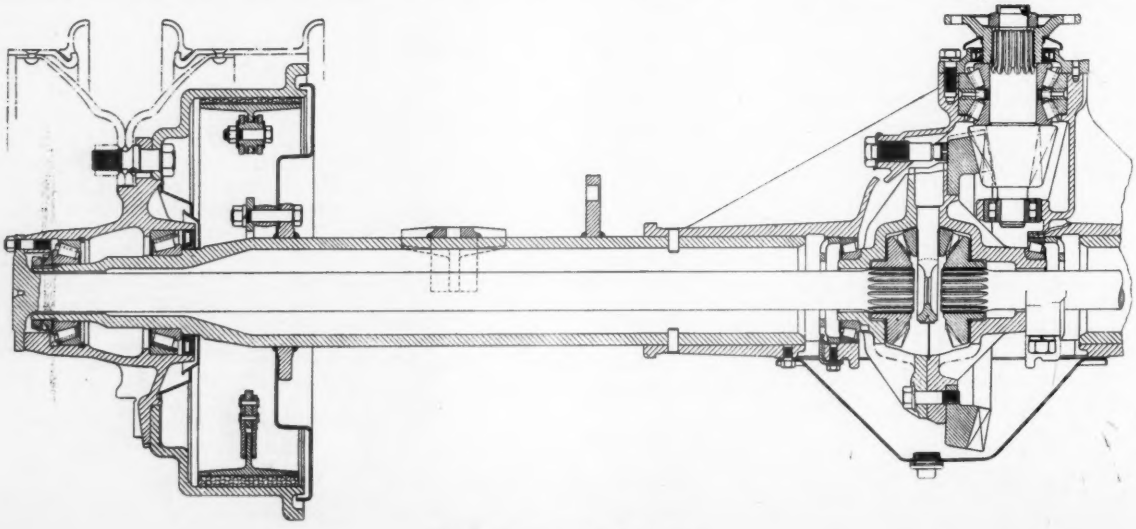
internally splined rear universal joint flange at the intermediate bearing. It bears against the double row ball bearing which is in turn located by a step on the circumference of the shaft, and the outer race is located in a malleable cast iron housing and secured by a circlip. Two lip-type oil seals are fitted, the front one working on the outside circumference of the propeller shaft and the rear one on the outside circumference of the universal joint flange. Both seals are protected from grit, etc. by large diameter sheet steel mud slingers. The housing is packed with grease during assembly and requires no lubrication during normal servicing of the vehicle. The universal joint spider and propeller shaft couplings are provided with grease nipples.

A thick circular section rubber ring supports the housing in a fabricated pressed steel bracket, and prevents transmission to the chassis of propeller shaft vibrations, whilst at the same time it allows a limited fore and aft movement of the housing relative to the bracket. A lug on the housing in conjunction with a rat-trap type spring retainer passing through holes in the bottom half of the bracket, locates the housing against rotational movement within the rubber ring. The bracket is bolted to the underside of the appropriate frame intermediate cross member.

Rear axle

A built-up assembly, consisting of a malleable cast iron differential carrier with pressed-in tubular extensions located positively by means of dowel pins, forms the axle housing. The brake anchor plates, shock absorber brackets and spring pads are welded to the tubes. The outer ends of the tubes are swaged down, and machined to carry the rear wheel taper roller bearings which are secured by two thin nuts per hub, and locked by a tab washer held between the nuts. On all models the centre of the differential carrier is on the chassis centre-line. A detachable pressed steel cover provided with a plug for oil replenishment is fitted on the rear face of the differential carrier. An oil drain plug is not provided, but the rear axle can be drained by removing the two bottom setscrews securing the gear cover to the axle housing.

Hypoid bevel gears are used for the final drive, with the pinion position $1\frac{5}{8}$ in. below the centre of the crown wheel, and $3\frac{17}{32}$ in. to the right of the chassis centre-line. The pinion is straddle mounted between a double row roller bearing at the inner end, and a double row taper roller bearing at the outer or forward end. A passage cast in the nose of the differential carrier provides for the cooling and



General arrangement of rear axle.

lubrication of this outer bearing as shown in the diagram.

Thrust in the three-pinion differential is taken by bronze washers behind each gear and pinion. The differential gears are floating in the malleable cast iron casing, thereby allowing equal loading of the three differential pinions. The two halves of the casing are assembled to the left-hand side of the crown wheel, which has tapped holes to receive the setscrews. These are locked by spring washers. A bronze thrust pad fitted to the left-hand side of the differential carrier opposite the pinion, provides an additional support for the crown wheel on heavy load. The assembly is mounted between two taper roller bearings housed in the differential carrier, and is located by threaded lock rings which are prevented from turning by a locking tab secured to the bearing housing cap by setscrews and spring washers. Thrust pads formed on the outer faces of the differential side bearing caps contact corresponding pads in the differential carrier. They prevent deflection of the bearing caps under heavy load. Contact between the pads is, of course, carefully checked on assembly at the factory.

The road wheel hub which is provided with an oil seal at its inner end is mounted on a pair of taper roller bearings. Location is provided at the inner end by a step in the outside diameter of the axle housing tube and at the outer end by two thin nuts between which is a lock washer keyed on to the axle tube. Lubrication is provided automatically from the centre of the axle.

The outer flanges which are an integral part of the half shaft forging are secured to the outer ends of the hubs by studs, nuts and spring washers. The inner splined ends of the shafts are carried in the differential side gears, and the two half shafts are interchangeable.

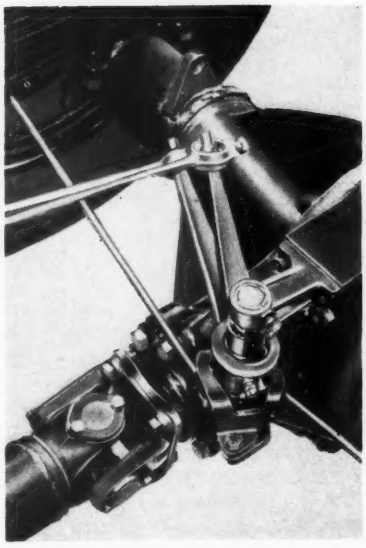
Without taking into account the various tyres and springs fitted to the individual vehicles, the designed maximum load which can be imposed on the tyres of the rear axle is 16,000 lb.

Axle Ratios

Models "SS" and "SL" 6·8 : 1 or 5·833 : 1.
Model "SA" 6·8 : 1.
Model "SB" 5·833 : 1 or 6·8 : 1.

Front suspension

The semi-elliptic springs are shackled at the rear end and secured to the axle by U bolts. A forward extension of the second leaf of all the front springs is turned up around, but clear of the spring eye, to prevent excessive rearward movement of the front axle in the case of a top leaf failure. The springs on models "SS" and "SL" are of the type which, by reason of a progressively changing load-deflection characteristic, reduce the tendency to a prolonged and violent bouncing effect due to resonance under certain conditions of repeated road impact. In addition, the progressive stiffening of the spring as the deflection increases helps to reduce the shock to the structure when full bump is struck.

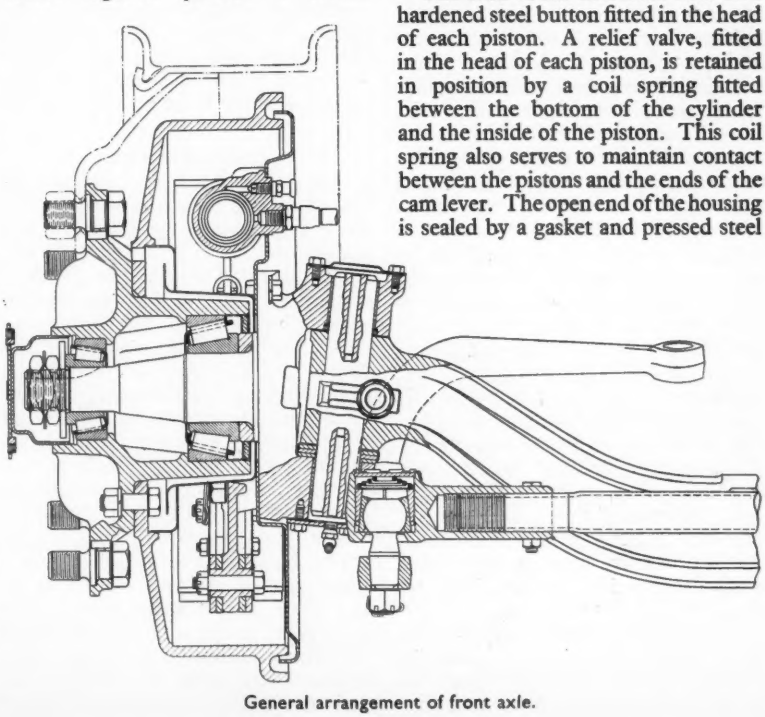


Brake operating levers.

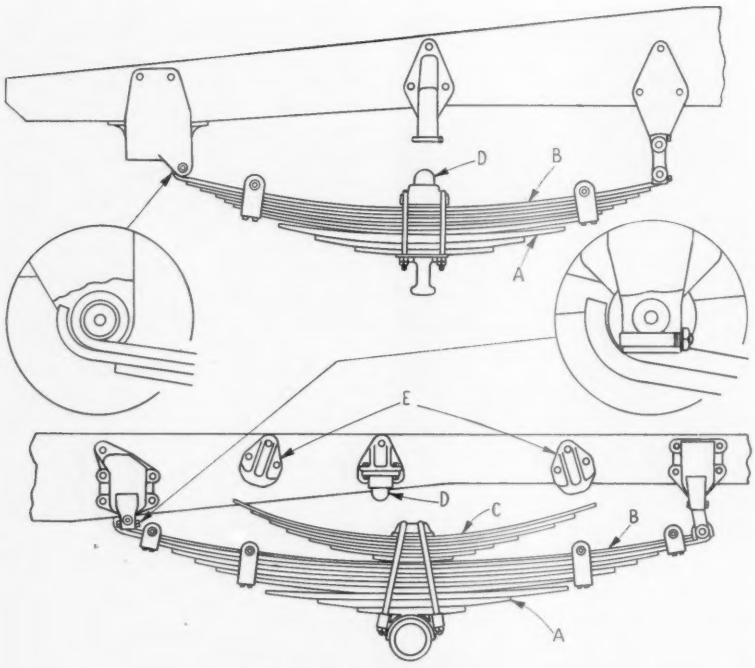
Model "SA" springs are fitted with tension plates which by binding the leaves together provide a certain amount of friction damping. The spring eyes and shackle brackets are fitted with rolled steel bushes, used in conjunction with chromium plated spring and shackle pins giving wide nominal running clearances in the spring eyes. It is claimed that this arrangement gives freedom of movement and long life, and these clearances should not be mistaken for excessive wear in a used vehicle. Shock absorbers are fitted as standard equipment on model "SB" and are available as optional extras on models "SS", "SL" and "SA", and are of the double acting hydraulic end to end discharge type. They are bolted to pressed steel brackets attached to the chassis frame side members. Operation is by a rubber bushed link connecting the shock absorber operating lever to a plate fitted between the front springs and front axle.

A cast iron housing, incorporating two vertical cylinders connected together at the closed bottom end by the valves and passages forms the shock absorber body, and a fluid reservoir is provided above the top of the cylinders. The cross shaft, located horizontally above the cylinders, carries a double-ended cam lever within the housing and an operating lever outside. The operating lever end of the cross shaft is provided with an oil seal carried in, and secured to, the housing by a pressed steel cup. The other end of the cross shaft bore in the housing is sealed with an expansion plug.

The ends of the cam lever bear on a hardened steel button fitted in the head of each piston. A relief valve, fitted in the head of each piston, is retained in position by a coil spring fitted between the bottom of the cylinder and the inside of the piston. This coil spring also serves to maintain contact between the pistons and the ends of the cam lever. The open end of the housing is sealed by a gasket and pressed steel



General arrangement of front axle.



Front and rear suspension.

cover secured by screws and spring washers. The filler plug is fitted in the cover.

A drop forging of I beam section is used for the front axle. The outer ends of the axle beam are set upwards and fitted with pivot pins which are inclined at an angle of 5 deg. 30 min. outwards from top to bottom. These pins carry the yoke-type steering knuckles. They are each secured in the axle beam by a large taper pin retained by a nut and spring washer.

Lead bronze steel backed bushes which have been proved on test to have exceptionally long life are fitted in the steering knuckles.—After 80,000 miles hard running the orginal machining marks were still visible on the bushes and wear was negligible.—The steering knuckle pivot bushes, and thrust bearings are lubricated by means of a nipple provided in the bottom cover of the steering knuckle, the lubricant for the top bearing passing up the centre of the drilled pivot pins. Lubrication of the top bushes is provided by a reservoir formed by the top of the pivot pins which project through the steering knuckle top bushes to give a stackpipe effect in the space between the top cover and the top face of the knuckle. A reservoir for the bottom bearings and thrust washers is provided by the hole up the centre of the pivot pins.

Taper roller bearings support the road wheel hub on the steering knuckle spindle. These bearings are secured to the spindle by two thin nuts, and locked by a tab washer sandwiched between the nuts, and keyed to the steering knuckle spindle. The camber angle is 0 deg. 45 min., and the toe-in $\frac{1}{8}$ in. to $\frac{3}{16}$ in. Castor angle varies in different models as follows :—

Model "SL" $\frac{1}{2}$ deg. negative.
Models "SS" and "SA" $1\frac{1}{2}$ deg. positive.
Model "SB" $\frac{1}{4}$ deg. negative.

Without taking into account the various tyres and springs fitted to individual vehicles, the designed maximum load which can be imposed upon the tyres of the front axle is 7,800 lb.

Rear suspension

The same device is used on the rear springs to prevent excessive rearward movement in the event of a top leaf failure, as on the front springs. At the rear end the shackles are likewise fitted with rolled steel bushes used in conjunction with chromium plated pins with wide nominal running clearances. This device, it is claimed, combines good wearing qualities with great freedom of movement. Referring to the diagram it can be seen that as the load, and therefore the deflection increases, a progressively greater proportion of the secondary spring A comes into contact with the primary spring B. Next the helper C comes into operation against the brackets on the frame, and finally the movement is arrested by the rubber bump stop D. Shock absorbers are supplied as standard equipment on model "SB" and are available as optional extras on models "SS" and "SL". The basic principles of the rear shock absorbers are similar to the front and need not be described again.

Brakes

The brake layout complies with the construction and use regulations for heavy motors cars. Hydraulic operation is applied to the front wheels and mechanical to the rear. Formed integrally with a reservoir for the brake fluid, the Bedford brake master cylinder houses two pistons in tandem. The front piston operates the front hydraulic brakes, and the rear one operates a lever located at the rear of the cylinder, which in turn actuates the rear mechanical brakes through a system of levers and rods. The tandem piston arrangement is a very desirable safety device, for in the event of a leak in the front system, or a mechanical failure in the rear system, the piston on the failed end comes up against a stop and prevents fluid escaping from the master cylinder, thereby keeping the good brake system in operation.

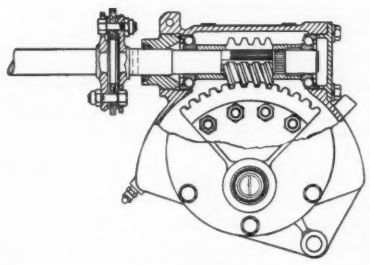
On models "SS", "SL" and "SB" a Clayton-Dewandre vacuum servo cylinder is mounted in combination with the brake master cylinder. The coupling between the two is by means of an internal linkage forming part of the combined assembly, and the vacuum master cylinder valve is connected by a pipe to the inlet manifold of the engine. The coupling between the servo and the brake master cylinder provides for direct connection between the two in the event of the servo becoming inoperative due to the engine stalling. Adequate feel-back to the controls is provided, and the assistance given by the servo is proportional to the pedal effort.

Shoe loading in the front brakes is automatically equalized by the hydraulic pressure in the cylinders located between each pair of shoes. These have separate pistons to actuate each shoe. Adjustment of the clearance between each brake shoe lining and the drum is provided for by means of a cam-type adjuster. A hexagon head, formed on the outer end of the adjuster, projects through the brake flange plate where it can be reached by a spanner. The anti-squeal device is a point worthy of special note on the front brakes. This consists of a spring-loaded tie at A on the illustration which causes the brake shoe flange to bear on two steel pads B, which are situated on the back plate near the ends of the shoes so that the shoe flange bears upon them. A straight line passing through their centres also passes through the centre of the spring tensioner. It is claimed that in most cases the friction damping effect, on

shoe vibrations, of this arrangement is sufficient to stop squeal.

Brake loading is equalized at the rear by means of the semi-floating operating lever situated on the differential carrier. The shoes in each brake are operated by means of this lever and tranverse rods coupled to bisector expander units located inside the brake flange plate. This wedge and roller type expander unit is designed to give automatic centralization of the brake shoes when acting in conjunction with the hinge links. A very good feature of the front and rear brakes is that the geometry of the hinge links (C in the illustration) and operating mechanism is arranged so that their resultant load acts at the centre of the shoe. This leads to more efficient braking, quicker bedding and more even wear than if the leading or trailing edge comes into contact with the drum, as is often the case with new brakes. Once the position of the levers is set by contact of the shoes with the drum, it is held by friction which is maintained in the linkage by the spring pressure plates (D in the illustration). Adjustment of the clearance between the shoe linings and the drum is provided for by a screw-type adjuster located between the operating wedge and the leading brake shoe in each brake. Access to this adjuster is through a hole in the brake flange plate.

The hand brake, which is of the horizontal pull-on type, operates the rear wheel brakes only. It is situated in the centre of the cab adjacent to the driver's seat. A cable connects the handbrake lever to the upper lever situated on the differential carrier.



Steering box.

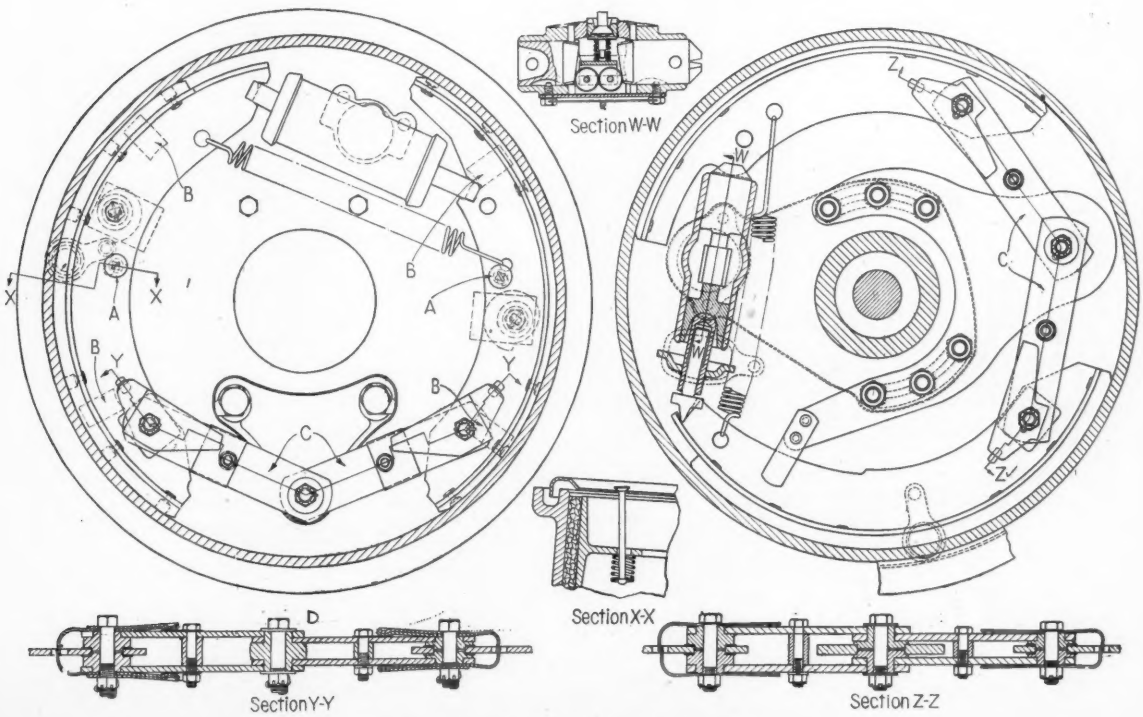
Steering mechanism

Semi-irreversible worm and sector steering gear is fitted. The malleable cast iron steering gearcase is rigidly bolted to the chassis frame side member, and the steering column is connected to the steering worm shaft by a fabric universal joint. This allows for slight malalignment and reduces the tendency for vibration to be transmitted to the wheel.

A hardened steel steering worm is located on its shaft by serrations. Distance pieces interposed at each side of the worm position the top and bottom combined radial thrust bearings, which support the worm. The bearings are adjusted by a threaded sleeve nut which screws into the steering gear casing, and is locked in the required position by a clamp bolt in the top of the casing. A felt oil seal working on the worm shaft is located in the top end of the bore of the threaded sleeve.

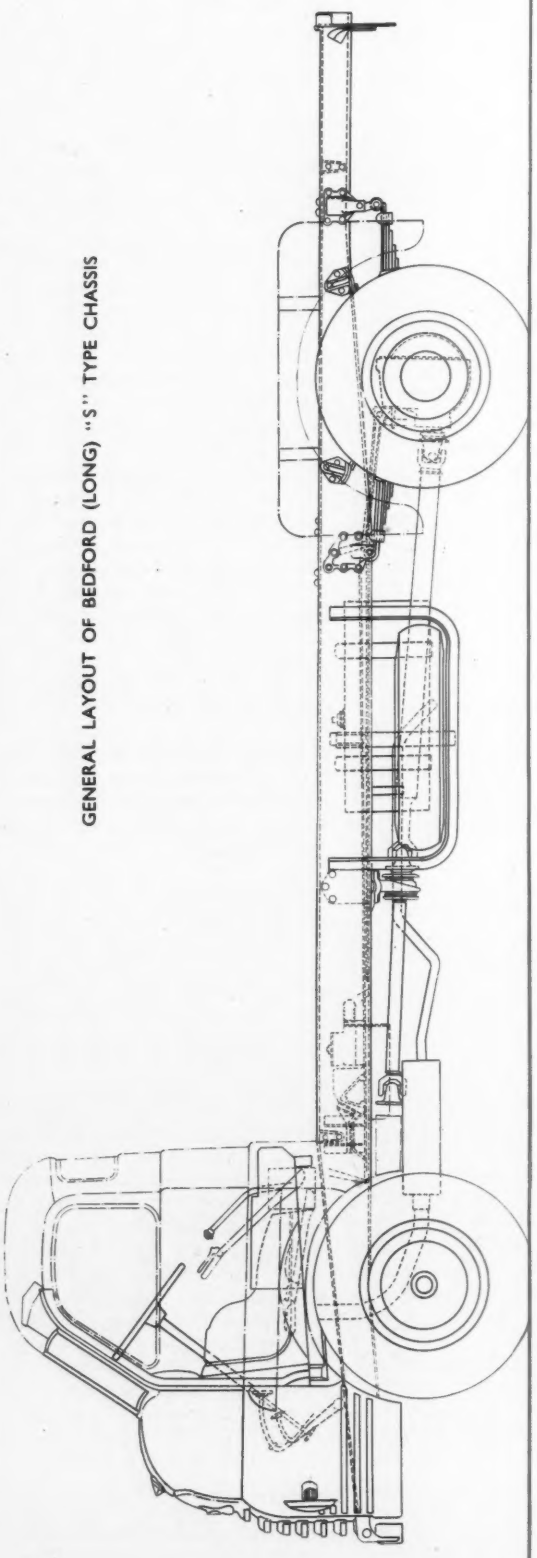
The case hardened nickel steel steering gear sector is bolted to the flange formed on the inner end of the drop arm shaft. Lead bronze steel backed bushes support the drop arm shaft in the malleable iron cover and which is spigoted and bolted to the steering gear case. End float of the drop arm shaft is adjusted by shims assembled between the cover and the case. Tapered serrations secure the drop arm to the shaft which is locked in position with a hexagon nut and heavy lock washer.

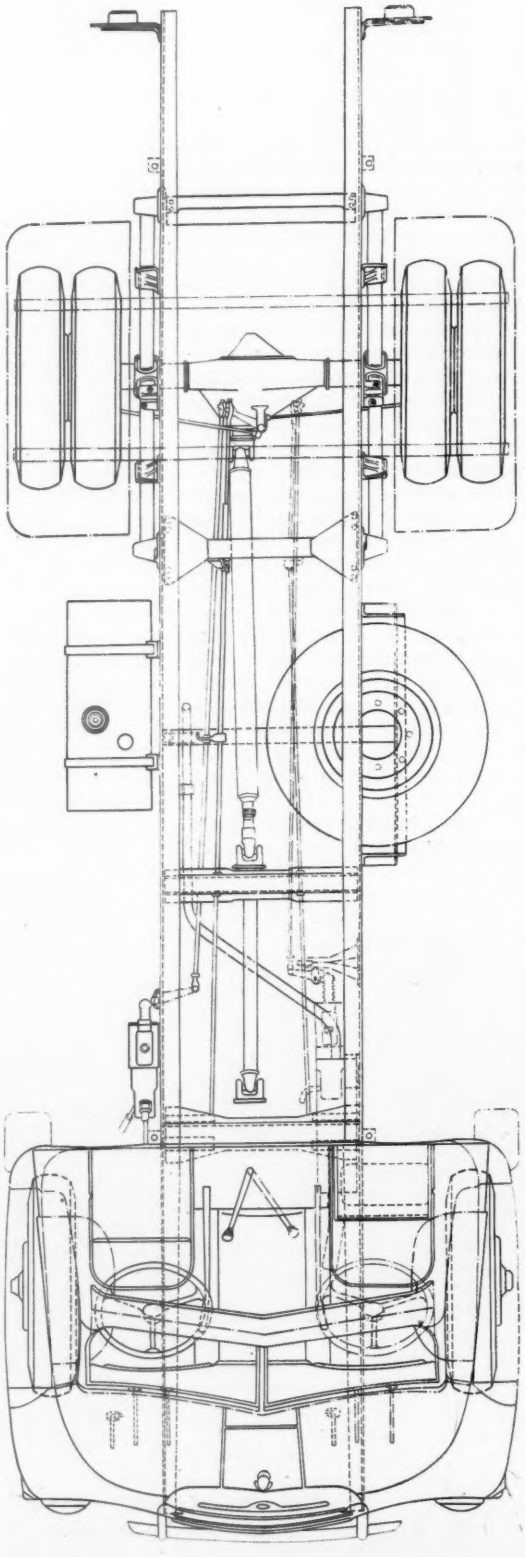
The steering column consists of a steel bar with the universal joint spider formed on the bottom end, and the top end is splined and formed for the attachment of the steering wheel. Spring loaded cup type ball bearings located at the top and bottom end of the cover respectively support the steering column in its short cover. These bearings are packed with grease during assembly. The cover tube is supported in a rubber ring arranged to roll in the pressed steel steering column support bracket attached to the dash under the instrument panel. This permits easy differential



General arrangement of front and rear brakes.

GENERAL LAYOUT OF BEDFORD (LONG) "S" TYPE CHASSIS





movement between the column and body, and insulates the column from vibration.

Hard moulded rubber over a steel core is used for the steering wheel which has two heavy spokes arranged at 128 deg. to each other in the bottom half, with $1\frac{7}{16}$ in. diameter stainless spoke in the centre of the top half. The horn push is housed in the hub of the wheel.

The forged steel non-adjustable steering wheel connecting rod is of I-beam section with wedge type ball joints at each end. The tubular steering tie rod is also fitted with these ball joints at each end, and the ends are threaded right hand and left hand respectively to provide adjustment for the correct alignment of the front wheels. A tie rod is located behind the front axle beam. Lubrication nipples are fitted to each steering ball joint and the steering box casing.

Frame

Pressed steel side members and cross member riveted together by cold squeeze rivets form the frame assembly. These rivets do not loosen by contraction after assembly, as in the case of hot formed rivets. The horizontal section of the front cross member is bolted to its side brackets and is detachable to allow easy removal of the power unit assembly. Towing eyes are fitted to the front of the frame side members except on model " SB." All models are provided with cross members at the front of the engine, the rear of the cab, and at the front of the rear springs. In addition, individual models are provided with the following cross members :—

Model "SS" at the back of the rear springs.

Model "SL" at the back of the rear springs, and one intermediate cross member between those at the rear of the cab and the front of the rear springs.

Model "SA" no additional cross members.

Model "SB" at the back of the rear springs, and two intermediate cross members between those at the rear of the cab and the front of the rear springs.

Frame data

Overall lengths of frame assemblies are :— Model "SS" 15 ft $6\frac{11}{16}$ in. Model "SL" 21 ft $11\frac{3}{16}$ in., Model "SA" 13 ft. $4\frac{5}{8}$ in.

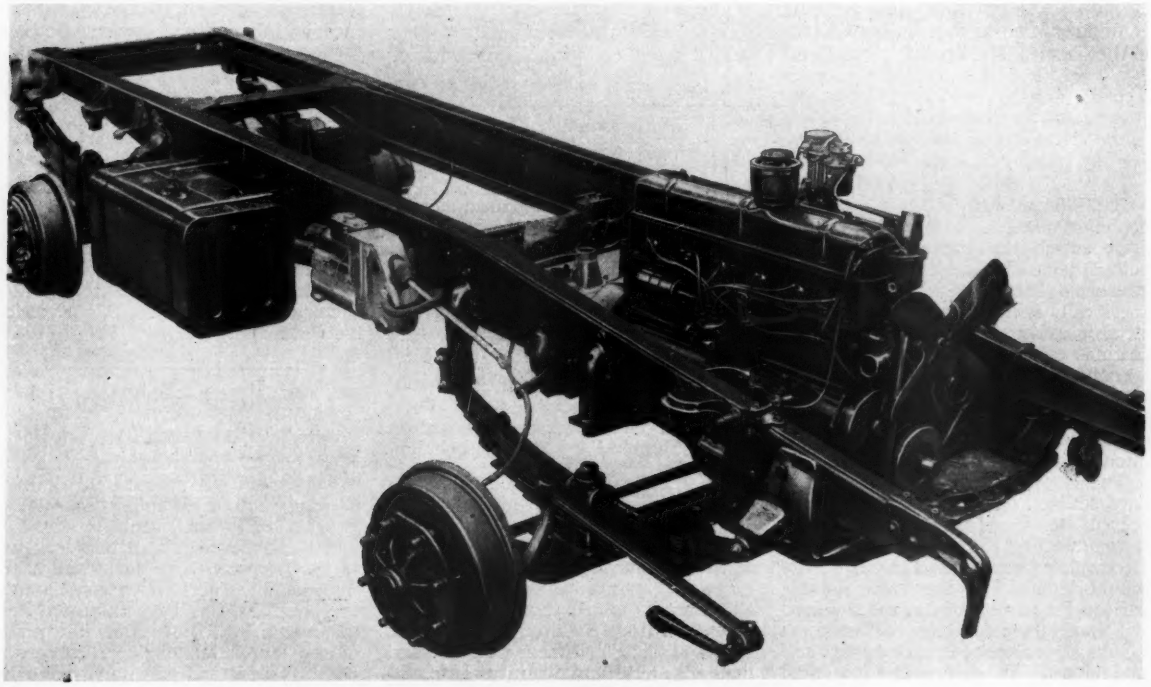
Models "SL", "SS" and "SA"	
Overall width frame assembly ..	3 ft. $11\frac{1}{2}$ in.
Width over side members, front ..	2 ft. $10\frac{3}{4}$ in.
Width over side members, rear ..	2 ft. $10\frac{3}{4}$ in.
Depth of main section of side member	9 in.
Width of side member flanges ..	3 in.
Thickness of side members ..	$(2\frac{1}{2}$ in. SA) $\frac{7}{32}$ in.

Electrical equipment

The electrical system is of the 12-volt positive earth return type, with single pole wiring. On Models "SS", "SL" and "SA", the dynamo output is regulated by the compensated voltage control system, and on Model "SB" it is regulated by current voltage system.

A Lucas distributor is mounted in an inclined position on the right-hand side of the engine. Automatic timing is controlled by a centrifugal governor, and a vacuum advance unit which supplements the centrifugal advance at part throttle in accordance with the depression in the inlet manifold. A scale for the manual setting of the distributor in the crankcase is incorporated on the distributor flange. With the distributor contact breaker points set to the recommended gap, the centrifugal and vacuum advances inoperative, the distributor points open 8 deg. before T.D.C., as shown by the bright steel ball embedded in the face of the flywheel, and the timing pointer. The bright steel ball and the timing pointer are intended to be used for timing purposes in conjunction with a Neon lamp operated from No. 1 cylinder and the engine idling below 600 r.p.m. The firing order is 1, 5, 3, 6, 2, 4.

The A.C. VF9 Ceramic insulator sparking plugs with end electrode setting are fitted on the right-hand side of the cylinder head. A Lucas B12-L/O ignition coil is mounted on the right-hand side of the engine. The dynamo is pivot mounted on the left-hand side of the engine and is driven



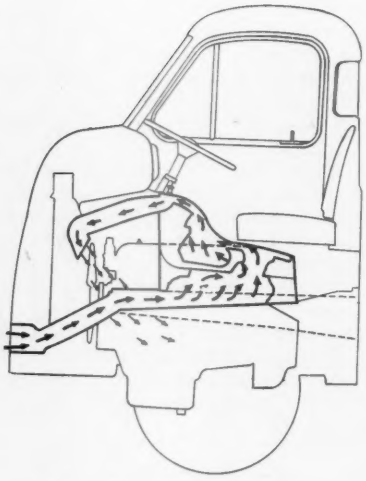
Engine installation in the Bedford "S" type chassis.

by a vee belt from the crankshaft pulley. Ventilation of the body of the dynamo is by means of a fan incorporated in the dynamo pulley.

The cab

The cab mounting is on three points, two widely spaced at the front, and one centre mounting at the rear. All mountings are of sandwiched rubber giving flexibility in shear in the horizontal plane. This, it is claimed, gives the frame freedom to distort whilst at the same time giving a firm vertical support to the cab. Access to the radiator is obtained by lifting the upper central section of the scuttle, which is hinged beneath the windscreen. Trap doors are provided in the floor of the cab for servicing work on the gearbox, mounting points, etc., and the toe panels are detachable.

A hinge at the rear end of engine cowl, and the removable side panels facilitate adjustments to the engine. The cowl is double skinned with an air gap between the inner and outer panels. A flow of air is maintained through the whole interior of the cowl by suction from the engine fan. The air intake is arranged behind the left-hand louvre at the bottom of the front cab panel, as shown in the diagram. It is claimed that this rather novel arrangement is extremely effective in keeping the cab cool in hot climates, and it is particularly necessary in view of the position of the cab over the engine with its hot exhaust manifolds. Additional cooling and cab ventilation is provided by four independent ventilators, two in foot wells and two in the scuttle. The driver's windscreen



Cab engine cover cooling system.

panel is hinged at the top and opens on adjusting quadrants.

Miscellaneous

A mechanical tyre pump is available as optional and extra equipment on all except Model SST, which is equipped with a hydraulic pump on the power take-off facing for the tipping gear. The mechanical tyre pump is mounted on the power take-off facing on the gearbox, and permanently piped to an air bottle mounted on the chassis frame. This equipment includes a flexible air pipe line and gauge assembly of ample length for attachment to the tyres.

The 20 gallon fuel tank on Model SS is made from tin-tern sheet pressings welded together, with the interior of the tank baffled to prevent excessive swirl of the fuel. A 26-gallon tank is available as an optional extra, for export only, on Model SS, but is standard on Models SL, SA and SB. On all models the attachment is by pressed steel side brackets and steel straps.

A straight through resonator type of silencer is fitted consisting of two concentric tubes supported in the outer shell by baffles. The concentric tubes and the baffles are suitably pierced to provide the silencing. Aluminium is sprayed on the outside of the assembly by the "Shori" process to increase resistance to corrosion. Two silencers in tandem are fitted on Model SB.

This chassis, as can be seen from the foregoing description, has many interesting features. The most important perhaps are those that are intended to reduce driver fatigue, and they include engine cover cooling, an opening windscreen, special anti-vibration devices on the steering mechanism, and the extra care taken with air intake silencing. These are additional to the more usual devices such as good seating and upholstery arrangements, good all-round vision, etc. Driver fatigue is a matter of great importance to the owners as well as to the drivers.

Great care has been taken in the design by detail attention to lubrication and sealing, as well as to the choice of bearing metal to ensure long life of the components. Sound engineering practice is followed, backed by experience, and many improvements on past designs are embodied.

BRITISH STANDARDS

*B.S.*1706. *Electroplated Coatings of Cadmium and Zinc on Iron and Steel* has just been published in the series which is being prepared covering electroplated coatings. The report in question deals specifically with protective coatings on electroplated articles. It does not cover coatings on sheet, strip or wire. The foreword points out that cadmium and zinc each have specialized fields of service and, therefore, the metal and class of coating should be chosen to suit the intended purpose.

The Committee preparing the standard realized that it was impossible at present to specify completely every factor affecting the performance of an electroplated coating but, in view of the urgent demand for some basis of comparison for such coatings, the standard is being published in its present form for the assistance of manufacturers and purchasers. It covers the essential qualities of the coatings in question, stipulating limits for the purity of the coating, the finish and appearance of the plated surface, the thickness and adhesion, together with appendices giving the methods of test. A section is also included dealing with the passivation of zinc coatings.

*B.S.*1718. *Tolerances for Steel Drop Forgings and Upset Forgings.* This Standard has been prepared at the request of the National Association of Drop Forgers and Stampers and is based on tables submitted by that Association. These agreed tables had been previously issued by the Association and are used by its members.

Tolerances are given in tabular form and are designed to cover the general range of forgings of average form in all ordinary specifications. Drop forgings are dealt with under thickness, width and length, draft angles, fillet and male edge radii, and quantity. Horizontally upset forgings are dealt with under thickness, die wear and shrinkage, eccentricity and off-set length bar portion and quantity.

Copies of these Standards may be obtained from the British Standards Institution, Sales Department, 24, Victoria Street, London, S.W.1., price 2/6d. each post free. (1961)

Welding and Metal Fabrication

READERS are reminded that the title of our associated journal *Welding* is now *Welding and Metal Fabrication.* The journal is wider in scope and covers all matters relating to the preparation of plate and sheet for welding processes. In consequence the page size has been increased and the journal will now cover the needs of the fabrication engineer concerned with all aspects of assembly, including metal cutting, pressing and bending before actual welding. (1937)

Simmonds Aerocessories Ltd. factory building.

NUT MANUFACTURE

A Survey of the Products and Production Methods of Simmonds Aerocessories Ltd.

DESPITE the degree of standardization that has been effected on thread forms, it is still necessary to produce a bewildering variety of nuts of different sizes, different thread forms and with different classes of finish. There are, in fact, 250,000 types of nuts in the standard products of Simmonds Aerocessories Ltd., Treforest, Glamorgan. So far as the automobile industry is concerned this organization is best known for the manufacture of Spire speed nuts and various types of self-locking nuts, and it is with these two types that these notes are chiefly concerned. There is also a large production of plain nuts.

The Spire speed nut, in all its many forms, is a remarkably interesting development that Simmonds Aerocessories Ltd. have introduced into this country. This type of fastening is based on the speed nuts originally developed by Tinnerman Products Inc., Cleveland, Ohio, U.S.A. There were well established in America before the war, but it is only in the past two or three years that they have been manufactured in this country. To-day, Simmonds Aerocessories Ltd. produce many speed nuts of their own design.

In a comparatively short time, Spire speed nuts have earned a good deal of recognition in many industries, including the automobile industry, despite the fact that they are completely different from any type of fastening used hitherto. There is no doubt that in the future, and as their special properties become more widely recognized they will be much more widely used.

Spire speed nuts

Although speed nut production has been carried on for only a relatively short time, Simmonds Aerocessories Ltd. already produce more than 120 different standard types and sizes. They vary greatly in form according to the application for which they are intended. In most forms the speed nut provides a double lock, that is, a compensating thread lock and a self-energizing spring lock.

The manner in which a Spire speed nut acts as a fastening device is illustrated in Fig. 1, which shows the form that was first developed and which still finds many fields of application. This illustration shows the pre-locked position at the top and the locked position below. On assembly of the nut to the bolt or screw, the two arched prongs move inward to engage and lock against the flanks of the screw thread. As these prongs have a relatively great range of movement they compensate for tolerance variations in the thread form. When the screw is tightened, the self-energizing lock is created by compression of the arch in the prongs and in the base. In combination these locks give positive protection against the danger that vibration will loosen the attachment.

It must be stressed that Spire speed nuts require only a low tightening torque, very much less, in fact, than is necessary with a conventional threaded nut. They must, of course, be tightened sufficiently to produce the thread lock and the spring lock. Any application of torque beyond the point necessary to bring these two locks into action is wasted effort. Indeed, if excessive torque is applied, there is a distinct possibility that the arch of the prongs may be distorted with a detrimental effect on their spring resiliency and on the efficiency of the speed nut.

As Spire speed nuts are so completely different from other types of fastening devices, it is worth while considering some of the advantages that may be gained from their use. These may be summarized as:—

(1) A Spire speed nut is easier to apply than an ordinary threaded nut, since it is a free-running fit on the bolt or screw threads and fewer turns are needed to tighten it.
(2) It is self-locking. Therefore, the need for a lock washer is eliminated.
(3) It has a wide surface that distributes the fastening load over a greater area than with an ordinary nut. As a result, a nut washer is unnecessary.
(4) Many forms can be used for blind assemblies in place of clinch or welded nuts, with appreciable savings in material and labour costs. In addition, if such assemblies are painted before the final assembly operations, threaded nuts

must be re-tapped, but no such necessity arises with a speed nut.

(5) There are forms, referred to as push-on fixes, that give an adequate lock on an untnreaded part.

(6) Because of their resiliency and the low tightening torque needed, Spire speed nuts can be used for the secure assembly of vitreous enamel, glass and plastic components without danger of damage in assembly or of vibration causing damage when the assembly is in service.

(7) Spire speed nuts, except the push-on fixes used with unthreaded parts, can be used over and over again without loss of efficiency.

Some conception of the savings that can be effected through the use of Spire speed nuts may be gathered from the results obtained on one application in an American automobile factory. For this application the speed nuts were used for mounting headlamps to wings. Before speed nuts were used, it has been necessary to weld threaded nuts to the wings for taking the headlamps.

When welded nuts were used, the wing pressings were transferred from the press shop and stacked in a storage area. Thence they were taken to the welding machines. Three welding machines, each with three operatives, were necessary to deal with the output. After the welding operations, the wings were transferred by hand to the metal-finishing conveyor line. Because of the welding and the number of times each wing was handled, a considerable amount of metal finishing was necessary.

At the end of the conveyor line the wings were Bonderized and painted and were then assembled to the car body. Before the headlamps could be attached, the nuts had to be re-tapped to clear the thread. Furthermore, from time to time a faulty nut would be found and the whole vehicle had to be side-tracked from the main assembly line for relatively costly rectification.

When speed nuts were used in place of the welded nuts, the metal-finishing conveyor line was extended directly to the press department. The welding and storage sections were no longer necessary and the floor area thus freed could be used for other productive purposes. From the press department the wings were placed in tandem, one right and one left, on the conveyor. Two operators, one on each side of the conveyor, snapped the self-retaining speed nuts into position as the wings passed their stations.

The direct saving in labour costs was considerable, since when welding was employed three skilled men and six helpers were needed to cope with the output, whereas when speed nuts were introduced into the assembly only two unskilled operatives were necessary. In addition, manual handling of the pressings was eliminated and much less metal finishing was required. Furthermore, if by any chance a speed nut proved to be faulty or if an operator had failed to attach one during the passage of the wing down the conveyor line, the need to side-track the vehicle no longer occurred. Instead, it was merely a matter of fixing a new speed nut in position. It is almost unnecessary to say that re-tapping was also eliminated. Somewhat similar results have been obtained in an English automobile factory.

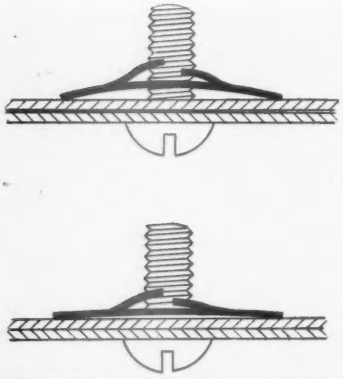


Fig. 1. A flat type Spire speed nut showing the pre-locked condition above and the double lock condition below.

It is not practicable to deal with all the many different forms and types of Spire speed nuts that are now in use, but consideration of only a few will serve to illustrate both the very wide range of applications for which they can be used successfully and the remarkably efficient production methods employed by Simmonds Aerocessories Ltd. The interesting feature of the production methods is that follow-on tools have been developed so that every type and form of speed nut is produced in a single press operation, despite the fact that many of them are of complex form.

All Spire speed nuts are made from coiled strip and all the presses are fitted with automatic feeding equipment and automatic means of ejection of the component. Incidentally, the strip is received in a standard width, but before it is mounted in the press, it is stripped to a blank width to suit the speed nuts that are to be produced. This allows the optimum use to be made of the material, an important factor in reducing material costs that has an added importance to-day because of the steel shortage.

In the production of the speed nut shown in Fig. 1, a central contoured hole is pierced by the first tool. The strip is then fed forward and on the next press stroke a pilot punch registers in the pierced hole. This ensures that all the strip in advance registers accurately under the appropriate tools. On the third press stroke, shearing tools split the strip longitudinally and for a controlled distance from the edge of the pierced hole. Two slits, equidistant from the centre line of the strip, are sheared on each side of the hole. At the fourth press stroke the prongs created by the shearing tools at the third stroke are formed into shape. Incidentally, it must be stressed that these prongs are accurately produced to suit the thread form with which they are to be used and when the component is finished their accuracy is tested against a conventional "go" and "not-go" thread gauge. On the fifth stroke the arch that gives the self-energizing lock is formed, and on the sixth and final stroke the speed nut is cropped off ready for heat treatment. A similar operation sequence is employed for push-on fixes that are used with unthreaded studs. A typical press set-up is shown in Fig. 2.

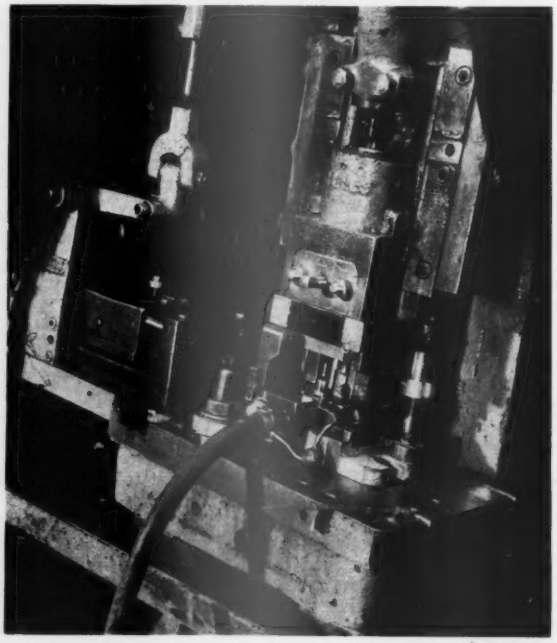


Fig. 2. Vertical power press and strip feed mechanism for the manufacture of Spire speed nuts.

So far as the automobile industry is concerned, the J- and U-type nuts shown in Figs. 3 and 4 are of great interest. They are particularly suitable for blind assembly applications, since they have self-retaining properties. Which type is used depends upon the nature of the application, but each eliminates the necessity for expensive riveting, welding or clinching operations. Among other applications U-type nuts have been used for headlamp assembly to wings, and J-type on the cowl and dash assembly of cab-over-engine vehicles and for the assembly of cab flooring for heavy commercial

vehicles. Each type can be installed to allow an appreciable misalignment of holes in the mating parts.

The J-type nut shown in Fig. 3 is produced on a vertical power press. At the first station two holes are pierced. One is circular. It is introduced to facilitate the bending that is performed at a later stage in the press sequence and the subsequent use in assembly. The other hole is contoured to suit the

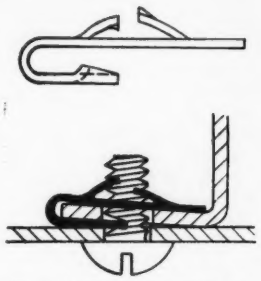


Fig. 3. J-type Spire speed nut.

thread form with which the nut will be used. When the strip is fed forward to the next station a pilot plunger registers in the circular hole while a D-hole is pierced and the shearing for the prongs is effected. Two tools operate at the next station also, one to form an extrusion round the D-hole, and the other to form the prongs. The next forward feed of the strip brings it clear of the guides in which it had been running and leaves it resting on top of a former placed transversely across the machine. On the next press stroke a tool that is recessed to pass over the

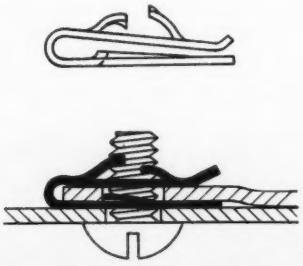


Fig. 4. U-type Spire speed nut.

former and which also carries a cropping tool comes down, crops the correct length and then forms the nut into a J-shape. A spring-loaded plunger in the recess holds the strip while the cropping is effected. Following this, the parted-off J-form is pushed along the former by a pusher operated by a cam slide to bring the nut under another tool which produces the final form. The type of tooling that is employed is shown in Figs. 5 and 6, but it should be pointed out that the illustration actually shows the tooling for one form of U-nut.

A Heenan and Froude four-slide horizontal press is used for the production of the U-type Spire speed nut shown in Fig. 4. In effect this machine comprises a blanking and piercing press

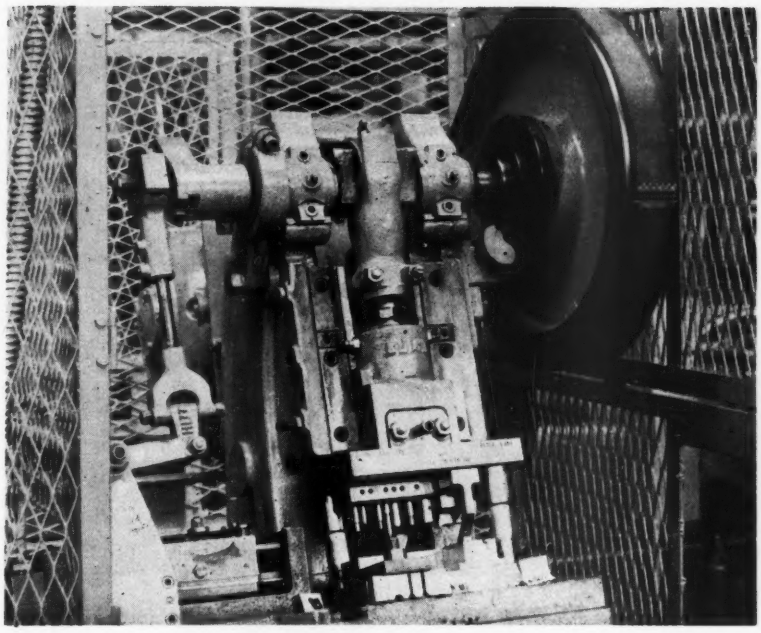


Fig. 5. A press tooled for the production of "U" type nuts.

followed by four slides which are used for forming the nut to the U-shape. It is illustrated in Figs. 7 and 8. At the first machine stroke two holes are pierced. One is a circular clearance hole through which the fixing screw will pass and the other a contoured hole to suit the desired prong form. On the second stroke a pilot plunger registers in the circular hole and a second circular hole is pierced. This hole facilitates the bending operation at a later stage and also simplifies attachment of the nut to the panel when assembly is being carried out. On the next stroke the shearing for the prongs is carried out, and on the final stroke before the strip leaves the blanking and piercing tools, the prongs are formed to suit the thread form and a locating tongue is sheared from the clearance hole.

The next forward feed of the strip brings it into position for forming from the four slides. When the advance of the strip is completed a pilot punch registers in one of the pierced holes in the next blank length, and the blank ready for forming rests in a shallow recess in the face of the front slide tool. This front slide then advances and grips the strip against a former. To give reaction against the push exerted on the former, the rear slide which is spring-loaded and cam-actuated, advances to contact the other end of the former. The blank length is then cropped off from the continuous strip and the front slide advances to start pressing the blank round the former. To complete the form, the right- and left-hand slides

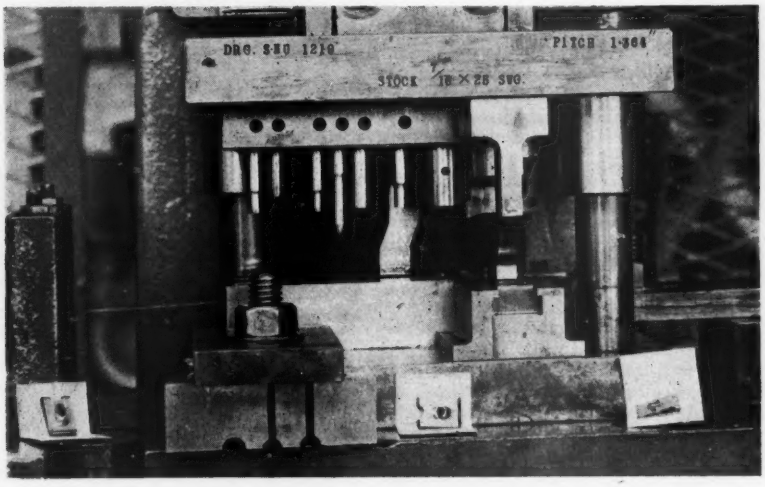


Fig. 6. The tooling in the press shown in Fig. 5.

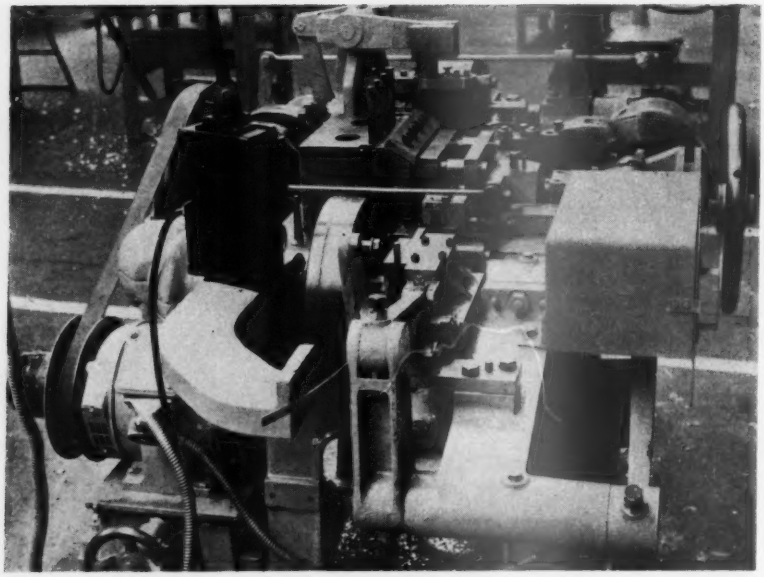


Fig. 7. Heenan and Froude four-slide horizontal press showing the feed mechanism and blanking tools for Spire speed nuts.

advance. When the slides have completed their strokes, the former is withdrawn and a kicker tool in the right-hand slide advances to obviate any tendency for spring effect to disturb the speed nut form. Finally the right- and left-hand slides are withdrawn and the completed pressing falls into the work container.

Another Spire speed nut that can be assembled, and if necessary removed, without there being access to the rear of the panel, is the latching type shown in Fig. 9. This nut was developed specifically for the assembly of wings to automobiles, but it is also suitable for many other applications. It is made from 21 s.w.g. spring steel and is designed for use with a special $\frac{1}{4}$ in., 12 t.p.i. Acme form screw. The ultimate tensile loading is 2,300 lb. and the recommended tightening torque is 80 in/lb., although a tightening torque of 120 in/lb. can be applied without impairing the efficiency of the nut. Because of the relatively high tightening torque, this nut is particularly suitable for power tool tightening. A saving of 20 minutes in assembly time per pair of wings has been effected through the use of this nut.

A vertical power press is used for producing the latching-type Spire speed nut. At the first feed of the strip two holes are pierced, one D-shape and the other contoured to suit the prong form. The straight side of the D-hole is the line across which the blank is eventually cropped off the continuous strip. At the second feed, a pilot punch registers in the contoured hole and the retaining tabs "A" are sheared. The shearing preparatory for prong forming is effected at the third stroke and at the fourth stroke the prongs and the retaining tabs "B" are formed. At the fifth stroke the latch "C" is sheared, a cropping tool cuts across in line with the straight side of the D-hole and the leading edge of the strip is formed up and is then bent over to the finished form at the sixth stroke. It will be noticed from Fig. 9 that this type of Spire speed nut has three prongs contacting the screw threads.

A very recent development of Spire speed nut for blind assembly is the O- or expansion-type illustrated in Fig. 10. It is suitable for many different applications where attachment must be made from one side only, and is particularly applicable where the fixing point is too far from the edge of the panel to permit the employment of J- or U-type nuts. In assembly this type of nut is first pressed into a square or round hole of appropriate dimensions and the spring arms retain the nut in the panel. If a flush face is required, the hole can be recessed. After the part to be attached is aligned, a coarse pitch screw is driven in. It forces the spring arms outwards. The friction fit of the screw between the spring arms makes this type of Spire speed nut very suitable for applications involving controlled adjustment of the screw.

In certain heavy duty applications it may be desirable to use a conventional threaded nut when blind assembly is necessary. Hitherto for such applications it has been necessary to use either a welded nut or a clinch nut, but Simmonds Aerocessories Ltd. have developed a Spire speed grip retainer, shown in Fig. 11, which greatly simplifies blind assembly work when a threaded nut must be used. A most complex set of follow-on tools is used in the production of this component with the nut in position.

To produce the speed grip retainer the strip is fed forward eleven times. At the first press stroke a circular hole is pierced. The second stroke is an idle one in which a pilot plunger registers in the circular hole. At the third stroke two longitudinal slots are pierced, and at the fourth one side of the form for the retaining lugs is blanked on each side of the centre. This blanking tool also produced the retaining lug blank form on the trailing edge of the blank next ahead. The next stroke of the press produces the second edge for the retaining lugs and at the same time the corner tabs are sheared. During the next three strokes of the press the retaining lugs are formed and at one of these strokes the corner tabs are also bent to form. When the form is practically complete, the nut is fed into position from a hopper and the retaining lugs are closed round it. The completed component is then cropped off.

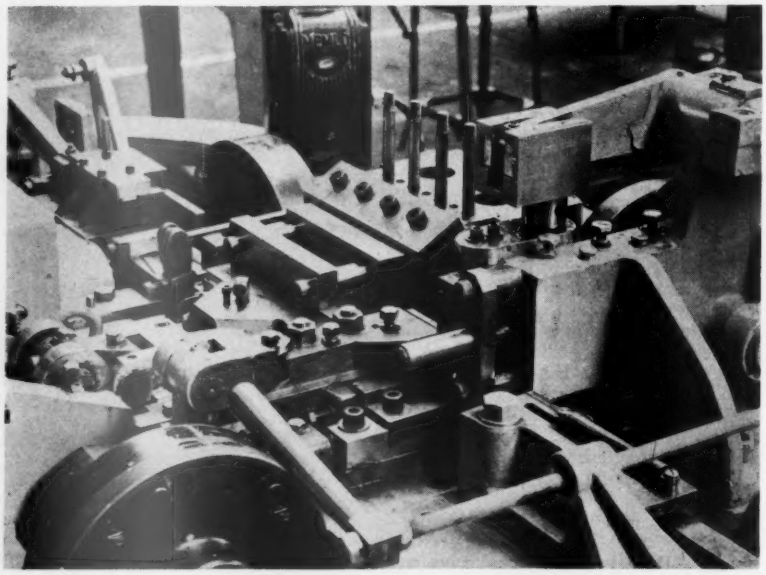


Fig. 8. The forming slides on the Heenan and Froude four-slide press.

Remarkably accurate follow-on tools are used in the production of every type of Spire speed nut, but probably the most complex form that is produced is the coil former support shown in Fig. 12. The following sequence is employed. At the first press stroke two holes with parallel sides and semi-circular ends are pierced. They lie across the strip, and in between them a longitudinal shape is sheared, from which the prongs are formed at a later stroke. At the second stroke two circular holes are pierced towards the edges of the

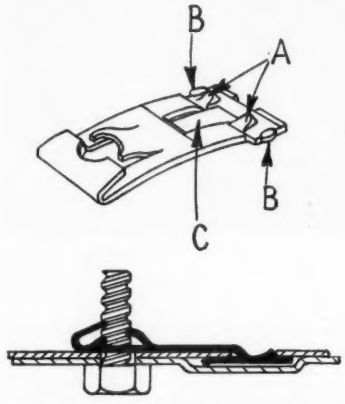


Fig. 9. Latching type Spire speed nut.

strip and in line with one of the holes pierced on the first stroke. At the same time, the metal sheared longitudinally at tne first press stroke is sheared transversely and a channel shape is also pierced. Next a dome and the thread helix are formed, following which at the fourth stroke, the periphery is blanked to form and the prongs are raised. At the fifth stroke the claws are formed and a tag is formed from the previously - pierced channel shape. Cropping off is effected at the sixth stroke and final forming at the seventh.

It is not possible within the compass of these notes to describe, even briefly, all the many different types of Spire speed nuts that have been developed. Reference may, however, be made to three types that are of interest to the automobile industry, latching type cable clips, tubular clips and knob clips. The latching type cable clip is designed to be snapped into a retaining hole where it will remain in position without bolts, nuts, rivets or welding. Tubular clips have been designed to provide a permanent lock for integral rivets or studs where there is access to only one side of the assembly. They are suitable for attaching emblems, name plates and other light assemblies. Knob clips have been designed for use with thermo-setting plastic components and die-castings. Their use eliminates the need for inserts or grub screws, reduces moulding costs and shortens assembly times.

The standard material for Spire speed nuts is spring steel which is finally hardened and tempered in the department shown in Fig. 13. Many speed nuts are supplied in the blued condition as received from heat treatment, and are Rodol dipped to give added protection against corrosion. If desired, the nuts can be given one of several anti-corrosive finishes, including cadmium plating, zinc plating, wax, zinc-chromate paint and Parkerizing. The barrel plating department is shown in Fig. 14. Although spring steel is the standard material, Spire speed nuts are also produced in stainless steel and phosphor-bronze.

Spire speed nuts are now used by practically all the leading automobile manufacturers. More than 300 of various types are used on one current popular model. The following list gives

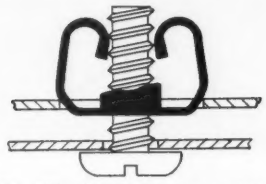


Fig. 10. Expansion type Spire speed nut.

some conception of the wide range of applications for which they are used:—

(1) Headlamps, tail lamps and roof lights, J- or U-types.
(2) Wing to body attachments, latching type.
(3) Radiator grille, U-type.
(4) Flooring, gearbox cover, luggage compartment floor, U-type.
(5) Trafficators, U-type.
(6) Instrument panel attachments, U-type.
(7) Emblems, badges and mouldings, push-on fixes.
(8) Interior trim, O- expansion-type.

Two applications are illustrated in Figs. 15 and 16.

In conclusion, it may be said that for suitable applications Spire speed nuts cost less than any alternative type of fastening. They also reduce working costs by simplifying assembly and shortening assembly times. At present they have a further advantage over conventional threaded nuts in that for a given number of attachments Spire speed nuts will use from 50 to 75 per cent less steel than the threaded nuts. As these nuts are completely different from any other type of fastening device, it is important that Simmonds Aerocessories Ltd. be consulted concerning their use while the project is still in the design stage. If this is done, the optimum results will be obtained.

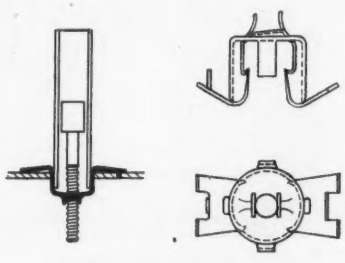
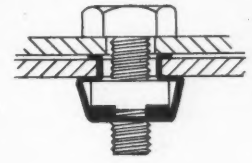


Fig. 12. Coil former support.

Threaded self-locking nuts

Simmonds Aerocessories Ltd. manufacture four distinct types of self-locking nuts, the Simmonds elastic stop nut, the Pinnacle nut, the Spunloc and the Nyloc. The Simmonds elastic stop nut which is shown in Fig. 17 is a standard hexagon nut with a counterbored extension that takes a fibre insert or collar. This insert has an internal diameter slightly smaller than the bolt diameter and is not initially



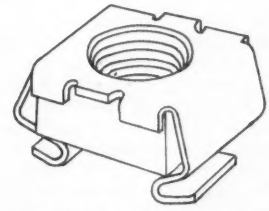


Fig. 11. Speed grip retainer.

threaded. It is securely anchored in the nut by a press operation that rolls the end of the nut over. When, in assembly, the bolt passes through the threaded portion of the nut and reaches the fibre insert, further movement is resisted. In overcoming this resistance, the nut is forced away from the bolt until the sides of the bolt and nut threads are in intimate contact. This force is distributed as a pressure on the sides of the bolt thread that will normally carry the load when the nut is fully tightened. As a result there is heavy friction between the threads of the bolt and the nut, with a consequent freedom from any possibility that vibration will cause the nut to come loose. As the bolt passes through the insert, it forms a thread in the fibre, but the elastic properties of the material are such that there is no danger that chips will be removed. Actually, the pressure between the bolt and the insert further increases the anti-rotational properties of this nut.

A later development is the Pinnacle nut shown in Fig. 18. This type has a metal diaphragm. It was developed for use in applications for which a fibre insert would be unsuitable, for example, where it is necessary to withstand elevated temperatures up to 500 deg C or where there will be contact with oils or other liquids which could act detrimentally on a fibre insert.

The diaphragn is made of special non-corrodable spring steel and is of a

Fig. 13. The heat treatment department.

diameter equal to the core diameter of the bolt with which it is to engage. It is de-pitched, so that when it engages with the bolt thread, it is deflected by an amount equal to at least half the pitch of the thread. This forces the nut away from the bolt to a degree that causes considerable pressure to be distributed on the load carrying side of the thread. As a result, there is maximum friction between the bolt and the nut. The diaphragm is so designed that the strain due to deflection is always within the limit of the proportionality of the material.

The most recent Simmonds development in self-locking nuts is the Nyloc. In principle, a Nyloc nut is similar to the original Simmonds elastic stop nut, but a nylon insert is used in place of a fibre insert. These nuts are suitable for a much wider range of applications than nuts with fibre inserts, since the nylon insert possesses better dimensional stability under varying conditions of temperature and humidity extremes, and is impervious to oils and boiling water. Within the range for which they are suited, Nyloc nuts are in fact better than an all-metal nut.

As nylon is still an unusual material for engineering applications, it is worth while to stress that the insert for the Simmonds Nyloc nut is produced from a special form of polymer with the following mechanical characteristics:—

Tensile strength ..	6,000-10,500 p.s.i.
Modulus of elasticity in tension	$3{\cdot}0\text{-}4{\cdot}0 \times 10^5$ p.s.i.
Elongation at break	30-35 per cent.
Cross breaking strength	12,000-15,000 p.s.i.
Density	1·14
Softening temperature ..	264 deg C.

This material is also strongly resistant to deterioration through contact with all oils, petrol, paraffin, kerosene and boiling water. Tests carried out in the Simmonds laboratories and at the R.A.E., Farnborough, show that the nylon insert is unaffected by solutions of hydrogen peroxide, 10 per cent soda ash, 5 per cent sulphuric acid, and 10 per cent sodium hydroxide at 85 deg C.

Fig. 14. The barrel plating shop.

In addition it is resilient, has a low coefficient of friction, and relatively great hardness that makes it strongly resistant to abrasion.

The efficiency of Nyloc nuts is demonstrated by the results of tests on six ¼ in. B.S.F. cadmium-plated steel nuts which were screwed completely on and off a standard bolt 30 times, and the average final unscrewing torque was 2·100 in/lb. Another nut of the same size was repeatedly screwed on and off a bolt and after 400 removals the torque figure was 1·00 in/lb. These results compare favourably with the aircraft specification A.D.114 requirement of 0·973 in/lb. after only 30 removals. An even later development is the Nyloc cap nut. The self-locking insert and cap are one integral moulding that provides a completely leak-proof seal. They are for applications such as pressurized chambers and fuel tanks.

The machining methods for all three types of self-locking nuts are similar. The blanks for the smallest sizes are produced on Brown and Sharpe single-spindle automatics, those for the medium sizes on multi-spindle bar automatics, and those for the larger sizes on semi-automatic capstan lathes. Only the method of producing nut blanks on multi-spindle automatics calls for comment.

For fibre insert and Nyloc nuts, six spindle machines are tooled to produce three nuts per machine cycle. At Nos. 1, 3 and 5 stations a circular form tool is mounted on the cross slide and a formed counterboring tool on the main longitudinal slide, while at Nos. 2, 4 and 6 stations a core drill of tapping size is mounted on the longitudinal slide and a parting-off tool on the cross slide. At the first station enough stock is fed through for three nuts plus three parting-off allowances. The tools at Nos. 1 and 2 stations machine the end of the bar remote from the collet, and the tools at the other stations are progressively nearer the collet so that when the cycle is completed three nuts have been formed on the outer surface,

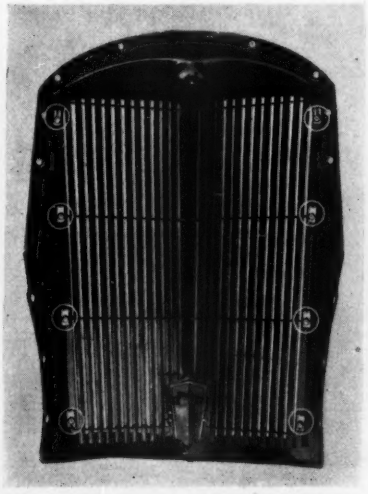


Fig. 16. Radiator grille for Mayflower.

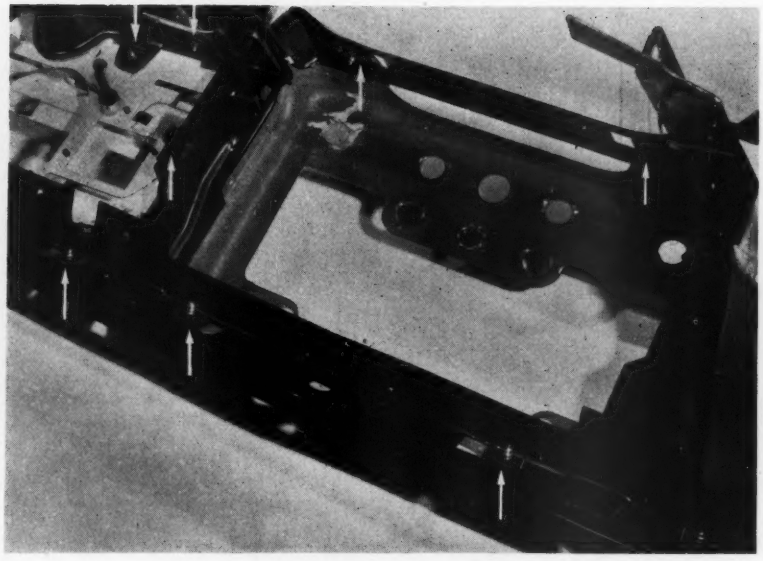


Fig. 15. Facia panel for Standard "Vanguard".

drilled for tapping and counterbored to take the inserts. A similar method is for Pinnacle nuts but only two nuts are produced per cycle. The reason is that the outer form for Pinnacle nuts is more critical, and if sufficient stock were advanced for three nuts there might be a danger of deflection and consequent lack of concentricity between the outer form and the counterbore at the first station, owing to the relatively long overhang from the collet. Part of the automatic section

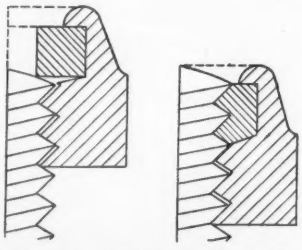


Fig. 17. A self-locking nut with either a fibre or a nylon insert.

for the production of blanks for self-locking nuts is shown in Fig. 19.

From the automatic section, the nuts are transferred to the tapping section where some are tapped on single-spindle conventional tapping machines, and others on automatic tappers with hopper feed. In themselves, the tapping methods are conventional, but it must be stressed that the greatest possible care is taken to ensure that the thread dimensions and form are correct. There is, of course, continual checking with thread gauges such as is carried out in any well-organized factory. In addition, when a new tap is used a sample of the work produced is sent immediately to the standards room. There, in addition to checking the thread with normal gauges, the thread is also projected.

Normally, the projection of an internal thread is a fairly lengthy procedure since the nut must first be carefully sectioned before projection is possible, and even then there is appreciable danger that the thread form may be damaged. Simmonds Aerocessories Ltd. have eliminated the necessity for sectioning the nut by using a special preparation that allows them to use the threads as a mould from which a casting can be produced. When heated, this preparation is sufficiently viscous

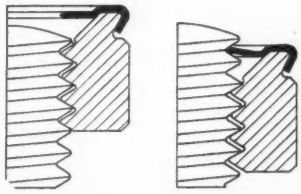


Fig. 18. Pinnacle self-locking nut with metal diaphragm.

to allow it to be impressed into the thread. It cools quickly without contraction and is ready for immediate projection. The thread form is not only examined under high magnification, see Fig. 21, but it is also projected on to sensitized paper so that a photograph is obtained and any errors in thread form can be shown to all concerned. This projection test is carried out at frequent intervals on random samples from all production.

The only other production operation that calls for comment is the securing of the inserts or diaphragms in the counterbore. For all self-locking nuts the insert or diaphragm is loaded into the counterbore and the end of the nut is then rolled over in a power press. On Nyloc nuts, the end is also keyed to the insert to give greater security. Generally these are successive operations, but

Fig. 19. A section of the automatic department.

recently some presses have been adapted for carrying out both operations at once on some of the smaller Nyloc nuts. One of these machines is shown in Fig. 20. An indexing turntable has been mounted on the press and is coupled up with the press mechanism in such a manner as to synchronize table indexing and the press stroke. The nut with the insert in position is loaded on to a pin and carried under the press ram where the tool both rolls the end of the nut over and also keys it to the insert. Unloading is effected in a most ingenious manner. At two stations beyond the working position, the pin on which the nut is loaded is automatically retracted and a blast of compressed air is directed on to the nut and blows it into a chute leading to the work container. The pin is then automatically advanced ready to take another nut.

Patrol inspection at every stage of manufacture and more detailed inspection after the machining has been completed in the automatic section, ensure that a high standard of quality is maintained. In addition to this, all self-locking nuts are inspected visually before they delivered to the warehouse. This inspection is carried out by girls sitting on either side of a conveyor belt, see Fig. 22. Each girl concentrates on only one item.

Regular tests are also carried out to determine that the tightening torque is in accordance with the specification and that repeated tightenings and removals do not affect the efficiency. Many of these Simmonds self-locking nuts must conform to A.I.D. specifications. As is well known, such specifications call for a higher standard than is necessary for normal commercial work. The form the tests must take is specified, and it is worth noting that for commercial self-locking nuts this organization carries out exactly the same test procedure as for aircraft work, although to slightly lower standards.

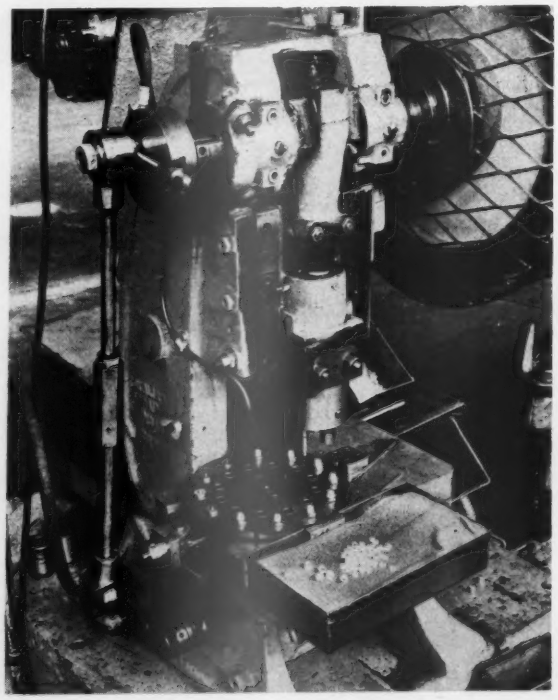


Fig. 20. A power press for securing and keying the nylon insert in Nyloc self-locking nuts.

Plain nuts

There is a large production of plain nuts, ranging in size from 6 B.A. to 2¼ in. B.S.F. The method of manufacture varies according to the size, although most of the blanks are turned in the automatic section on single-spindle or multi-spindle automatics. At first sight the use of multi-spindle automatics for the production of a relatively simple component such as a plain nut may seem wasteful, but in fact it gives a very high production rate since no less than five nuts are produced per machine cycle.

Certain sizes of plain nuts are also produced on Waterbury-Farrel cold heading machines. This is probably the fastest available method of producing high quality plain nuts. The nuts produced on these machines are tapped in an automatic tapper with hopper feed. Certain sizes of bolts and screws are also produced in the cold heading department. For these, National Boltmaker machines are used. They do not include the flat

Fig. 21. Checking a nut thread form by projection.

Fig. 22. Final inspection on self-locking nuts.

rolls as an integral part of the machine, and the thread rolling is carried out in a separate machine.

Important though it is, only brief reference can be made to the work of the inspection department. In general, this department is organized on conventional lines for dealing with every aspect of production from the receipt of raw materials to the despatch of the finished articles. It is worth noting, however, that the system of patrol inspection does not relieve the machine operators from responsibility for the quality of the work produced. This is a factor that reduces unproductive labour costs without lowering the quality of the products. In fact, since the operators have had to carry some responsibility for their work, there has been a reduction in the number of parts rejected for failing to conform to the specified standard. In addition to the usual inspection functions carried out in the production departments there is a large and well-equipped standards room which is recognized as an A.I.D. approved test house.

Hydrodynamic Drive Terminology

RECOMMENDATIONS of an S.A.E. committee on standard terminology for torque converters and fluid couplings have been published recently. The importance of standardization at this early stage of development cannot be over emphasized. Many papers have been, and will be published on hydrodynamic drives on both sides of the Atlantic, and it is most essential that these papers can be easily followed, and that no confusion shall be caused by differences of terminology.

It is, for instance, necessary to be absolutely clear about the distinction between the "number of stages," and the "number of phases" in a converter. The term "number of stages," as applied to a torque converter, refers to the number of driven members separated from each other by members of different function, such as stators for input members. Whereas the expression "number of phases" is defined as the number of functional arrangements of the working members when the functional change is by mechanical means, for example, a stator which may free wheel in one direction. It is interesting to note that the expression "hydrodynamic drive" is a collective term applied to all drives transmitting power solely by dynamic fluid action, and therefore includes both torque converters and fluid couplings.

All engineers and others who are interested in the subject, and who are not absolutely clear about the meanings of such terms as "bias," "scroll," "racing speed," "mean camber line," etc., should certainly refer to the recommendations which are published in the *S.A.E. Journal*, March 1951. Whilst it does not seem proper that the British, who are the originators of the English language, should turn to America for their terminology, it does, however, seem highly desirable to standardize in this subject, and it is to be hoped that the British Standards Institution will take the matter up further in co-operation with both the S.A.E. and British industry. (1952)

Fox Steels

A BOOKLET dealing with Fox steels has recently been issued by the United Steel Companies Limited, 17, Westbourne Road, Sheffield, 10, in which the specifications of the alloy and special steels manufactured by Samuel Fox & Co. Ltd. are concisely tabulated. In addition to the standard qualities of Fox brands, specifications are given of the British "En" series, "S" series aircraft specifications, "DTD" series standard specifications, A.I.S.I. and S.A.E. open hearth and electric furnace alloy steels, and A.I.S.I. stainless and heat resisting irons and steels. The publication also embodies a number of useful tables such as standard wire gauges, British and metric load equivalents, hardness and tensile strength equivalents, conversion factors, etc. (1939)

Rolls-Royce Ltd.

THE Board of Directors of Rolls-Royce Ltd. announce that Mr. A. G. Elliott, C.B.E., has been appointed joint managing director of the Company with Lord Hives, C.H.

Mr. Elliott, one of the original team of Napier designers, joined Rolls-Royce in 1912, working directly with the late Sir Henry Royce on engine design for more than twenty years. In 1937 he was made chief engineer of the company and was appointed to the Board in March 1945.

NEW PLANT AND TOOLS

Recent Developments in Production Equipment

A SPECIAL - PURPOSE machine developed by Adcock and Shipley Ltd., Ash Street, Leicester, is illustrated in Fig. 1. It is a 14-spindle horizontal automatic cam feed drill for work on motor vehicle shock absorbers. As with other Adcock and Shipley special-purpose machines, this machine incorporates a unit head on to which multi-spindle drill heads are fitted. The model 3S unit head fitted to this machine is a new design and is larger than those hitherto manufactured.

There are eight stations in the fixture, one for loading and unloading and seven working stations. The components are loaded in pairs and the spindles operate in pairs to carry the following sequence of operations on the relief valve holes of the shock absorbers: —

No. 1 spindle drills a $\frac{1}{16}$ in. diameter hole.

No. 2 spindle drills a $\frac{1}{4}$ in. diameter hole.

No. 3 spindle reams a 0·280 in. diameter hole.

No. 4 spindle faces the bottom of a $\frac{1}{16}$ in. diameter hole.

No. 5 spindle chamfers the 0·280 in. diameter hole.

No. 6 spindle recesses a 0·643 in. diameter hole.

No. 7 spindle recesses a $\frac{11}{16}$ in. diameter hole.

Because of the arrangement of the holes it is necessary to carry out two operations on each component. Therefore the method of operation is: —

(1) The components are loaded at No. 8 station and then go through the sequence of operations outlined above.

(2) They are removed at No. 8 station and turned round to allow the same operation sequence to be carried out on the other hole.

(3) The components are unloaded and fresh unmachined components are loaded in to the fixture. This fixture is machined so that each station is inclined at the correct angle to bring the relief valve holes horizontal.

Essentially the machine comprises a suitable base, fabricated from steel plates, carrying on the right hand side two vertical castings between which the hand-indexing fixture is carried on a very large diameter ball-bearing trunnion. The unit head is mounted on a sub-base on the left hand side of the bed. This No. 3S unit head is fitted with a 7½ h.p. flange-mounted motor which operates the feed through a barrel-type cam and also drives the 14-spindle multi-head that is fitted to the front of the unit head.

The barrel-type cam feed gives quick approach, change to feed rate and then rapid return to the rear position, where the head stops automatically. Changes of feed rate are effected by means of pick-off gears conveniently situated under a cover on the head. In addition, a hand feed is provided for try-out purposes. A sub-base carries the unit head. It can be moved along the main bed, to give clearance for tool changing, by a screw and nut operated through spiral gears by a hand lever. To ensure rigidity, the sub-base can be locked in any desired position.

Very robust construction is used for the 14-spindle multi-head, which is fitted with ball-bearings throughout. The spindles are bored to take drill adjusters for individual adjustment of the tools. Lubrication to all gears and bearings in the head is effected from a pump through a visible oil flow indicator and distribution box.

The work fixture, carried between the two vertical castings, is fitted with quick-action clamps and pegs for accurately locating the components. Both vertical castings carrying the fixture are of heavy section. The rear one acts solely as a bearing support for the fixture, while the front casting, in addition to acting as a bearing support, also acts as a guide plate carrying interchangeable hardened guide bushes. The fixture is located after indexing by a taper locating plunger which is retracted for indexing through a conveniently-placed lever. This machine has a time cycle of 24 seconds.

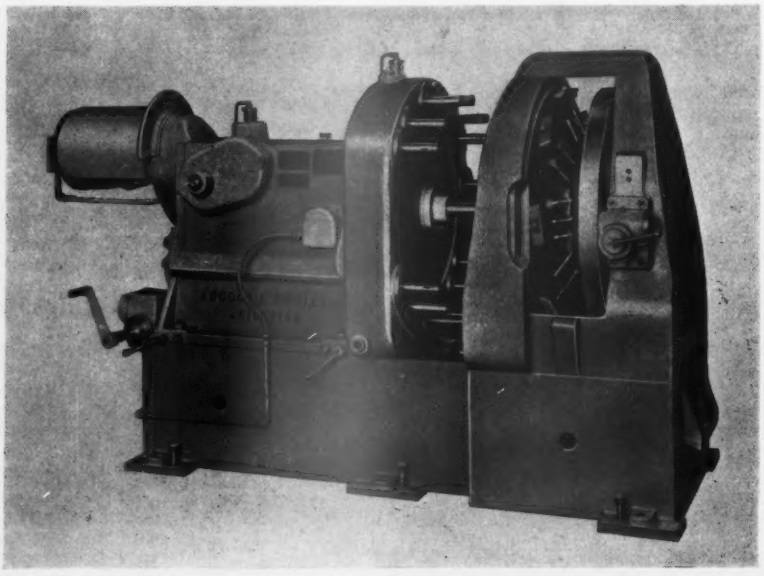


Fig. 1. Adcock and Shipley 14-spindle horizontal cam-feed drill.

Single spindle automatics

A completely re-designed single spindle automatic machine, illustrated in Fig. 2, has recently been developed by B.S.A. Tools Ltd., Mackadown Lane, Marston Green, Birmingham. Three sizes of two types are available. One type, designated "AL," has a long stroke turret, and the other, "BL," has a short-stroke turret. The sizes are: —

No. 98—1⅛ in. diameter maximum capacity, types AL and BL.

No. 138—1⅜ in. diameter maximum capacity, types AL and BL.

No. 168—2 in. diameter maximum capacity, type AL only.

AL type machines have two automatic changes of speeds in forward and reverse, that is, a total of four spindle speeds in one cycle. BL machines have one automatic change of speed forward and reverse, or alternatively two speeds forward or two speeds in reverse.

In the design of these machines special attention has been paid to the problem of swarf removal. Actually, the swarf drops straight down to a tray, which can be emptied from the right hand end of the machine during operation. It does not interfere in any way with tool setting or loading. An automatic conveyor can be fitted in place of the tray, in which case the swarf is deposited in the top half of a special container while the oil drops through a strainer into the base.

A wide range of speeds in forward

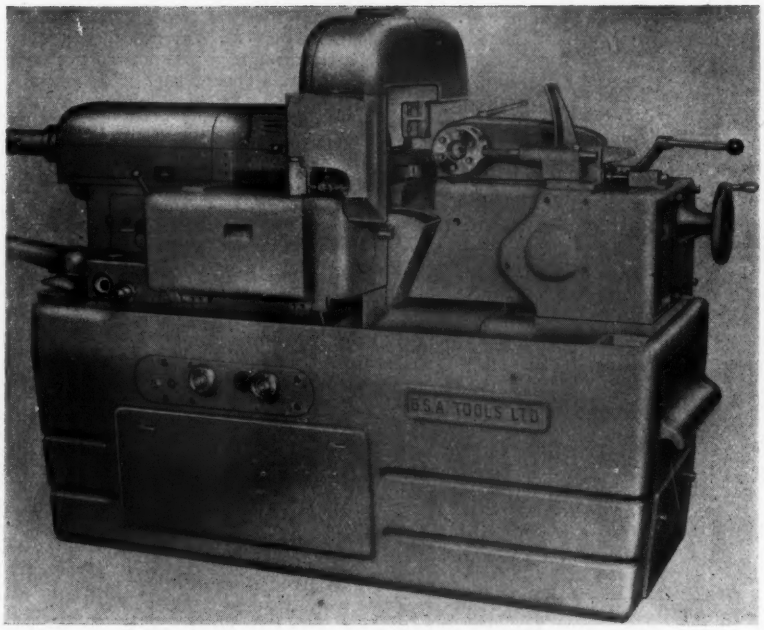


Fig. 2. Type 'L' B.S.A. Tools Ltd. single-spindle automatic machine.

and reverse directions can be obtained by means of pick-off gears. All shafts in the gearbox are mounted on ball-bearings and are provided with automatic lubrication. The front end of the workspindle on No. 98 machine is mounted on roller bearings, while the workspindles of Nos. 138 and 168 have taper bronze bearings with micrometer adjustment for taking up wear without affecting the concentricity of the spindle. On all three sizes the thrust is taken by opposed angular contact bearings located at the rear end of the spindle. Provision is made for adjustment to eliminate play.

Drive from the gearbox to the spindle is by duplex roller chain through two cone-type clutches. This gives a smooth and positive drive and allows the spindle bearings to be brought close together. Jockey sprockets, controlled by two knobs on the front of the machine, maintain the correct chain tension.

The totally enclosed backshaft is driven at constant speed by a belt from the main drive shaft. Engagement is effected by means of a lever-operated clutch. Tripping of the various clutches is by means of push rods that pass through the body of the machine. This arrangement dispenses with the necessity of large openings in the body. An overload device is fitted for knocking out the backshaft starting clutch. If a transfer unit is used, any obstruction to the transfer arm operates the knock-out device.

Cross slide movements, stock feed, spindle speed changes and reversal, and turret cam drive are all controlled by the front and turret camshafts. Although they are totally enclosed, these mechanisms are easily accessible when their hinged covers are opened. They are driven from the backshaft through a worm drive and pick-off gears that give 106 different cycle times which are completely independent of spindle speeds. The gearbox for obtaining cycle times is a constant centre type.

The turret, which has six tool stations, is constructed to give rigidity to the turret tools, so that heavy turning cuts can be taken. It moves on large square section ways and can be adjusted forwards or backwards in relation to the spindle. Turret slide feed is actuated by a cam-operated quadrant and rack. Indexing is effected smoothly and rapidly by means of a link and Geneva motion mechanism.

A positive feed pump lubricates the work spindle bearings and the cross slides. The gearbox bearings, chains and gears are supplied with lubricant from a mechanical pump. All other bearings are lubricated either by a "one-shot" pump or by separate oil wells. An ample supply of coolant is provided by a coolant pump and fittings that are included as standard equipment.

The bar feed and collet opening mechanism is actuated by trip dogs controlled by a trip drum on the front shaft. The length of the bar feed can be adjusted up to a maximum of 4 in. on the long stroke and 2½ in. on the short stroke machines. All the moving parts of the machine, other than the work spindle, are stopped automatically when the end of the bar stock is reached. This is effected by a device that disengages the back shaft. Two adjustable trip dogs on the front camshaft operate the work chute in such a manner that swarf is effectively separated from the work. The timing of opening and closing can be varied without the use of special cams. Particulars of spindle speeds and cycle times are:—

No. 98AL, 79 speeds from 42 to 2840 r.p.m.

No. 98BL, 106 speeds from 24 to 3135 r.p.m.

No. 138AL, 88 speeds from 42 to 1850 r.p.m.

No. 138BL, 84 speeds from 25 to 2230 r.p.m.

No. 168AL, 70 speeds from 43 to 1260 r.p.m.

Cycle times, long stroke machines, 6-700 sec., short stroke machines, 3-350 sec.

Bearing cap screw tightener

A machine that has been developed by H. E. Farmer Engineering Co. Inc., Detroit, U.S.A., for tightening cap screws for securing main bearing caps to the cylinder block is illustrated in Fig. 3. It is designed to give a balanced tightening torque of 85 ft/lb., plus or minus one lb., and both screws for one cap are tightened simultaneously. There are two working heads each driven by its own motor. Drive is taken from the motor to the two spindles through a worm wheel and differential. A magnetic clutch is inserted in the worm shaft and a Kirkman torque control unit is wired into the motor line. As soon as the required torque is reached the clutch opens and the spindles stop rotating.

This machine has been developed for use on a specific cylinder block with

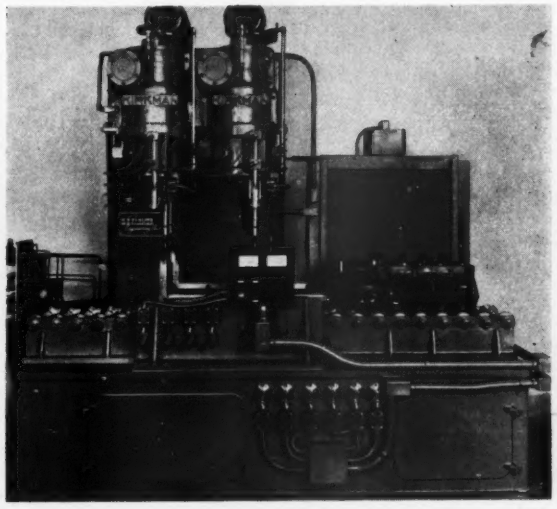


Fig. 3. Bearing cap screw tightener.

five bearings. The block is pushed into the machine with the clutch end leading until it reaches a stop which brings No. 4 bearing into position under No. 1 head. Depression of a push button then starts the following operation cycle:—

(1) The block is hydraulically clamped.

(2) The quills in No. 1 head advance in rapid traverse, engage the cap screws and apply the specified tightening torque.

(3) The quills are retracted, the clamps are released and the first stop is retracted.

The block is then advanced to No. 2 stop which positions No. 3 bearing cap under No. 1 head, and the cycle is repeated until No. 1 bearing cap is in position under No. 1 head. At this stage, No. 5 bearing cap is under No. 2 head and the cap screws for Nos. 1 and 5 bearings are tightened simultaneously. A second head dealing only with one bearing is necessary because the cap screws for No. 5 bearing are offset in relation to those of the other bearings. This machine, the first of its kind, will deal with 42 blocks per hour. A larger model with more heads is being produced to deal with 135 blocks per hour. The agents for this machine are Gaston E. Marbaix Ltd., Devonshire House, Vicarage Crescent, London, S.W.11.

Surface projector

A recent addition to the range of measuring equipment designed and manufactured by Optical Measuring Tools Ltd., Slough, Bucks, is the surface projector shown in Fig. 4. It has been designed for the projection and measurement of surface contours, such as type faces and matrix punches, where extremely fine dimensions and form control are essential. To cater for a large range of magnifications, this instrument incorporates infinitely variable magnification over the range 30× to 180×. This range is divided into five parts by the use of five specially designed lens systems mounted on a turret that can be rapidly and accurately brought to the desired position. Each lens covers part of the magnification range. Any value of magnification within the range of the lens system in use can be selected by operating a push button control that operates an electrical mechanism for changing the position of a mirror system within the projector, thereby increasing or decreasing the magnification.

Fig. 4. O.M.T. surface projector.

The instrument control panel, conveniently grouped for ease of operation, incorporates a window through which five engraved drums are observed. Each drum is connected to its appropriate lens and is marked with a magnification scale. When the push button control is operated the drums revolve slowly so that any desired magnification within the range can be selected. Brilliant illumination is provided at all magnifications by the lighting unit and condenser system, which are arranged to require the minimum of manual adjustment.

For the rapid alignment of the work image with the screen drawing, the work stage includes a fine focusing control and co-ordinate movements. The plate of the work stage is designed to accept suitable fixtures for mounting work pieces. The opaque screen is positioned at a convenient height for comfortable observation. It allows drawings to be made on metal to avoid the serious dimensional changes that occur with paper and plastic materials used with translucent screens. There is an effective screen area of 24 in. × 24 in.

Synthetic Resin Adhesives

GOOD gluing depends not only upon good glues but also upon correct methods of using them. Specialized experienced on the part of chemists, physicists and engineers is called for in the making of synthetic resin adhesives, and a thorough knowledge of gluing technology is also required of technicians whose responsibility is to control their use. Only when the experience of the manufacturer is allied to the skill of the user can the advantages of modern adhesives be exploited with complete success.

These reflections have prompted Aero Research Ltd. to organize a course to embrace most aspects of the use of synthetic resin adhesives. Adhesives of this class now form an integral part of plywood as well as of furniture, musical instruments, motor cars, aircraft, sports goods, electrical components, shop fittings, buildings, ships, boats and many other finished products.

Course (1) is mainly for the woodworking industries, but Course (2) will be of interest to automobile engineers. This course will deal primarily with adhesives for metals and non-porous materials. It will include instruction on preparation of surfaces for bonding (e.g. pickling, degreasing), recommended joints, methods of testing, effects of surface treatments upon joint strength and electrical properties of adhesives. The fee for the Course is £10 10s. 0d. inclusive of accommodation, meals and transport. Applications for enrolment and full particulars should be made to Aero Research Ltd., Duxford, Cambridge. (1953)

Trico Accessories

THERE were many interesting items on the Trico-Folberth stand at the Castle Bromwich Section of the British Industries Fair. Among these was a headlight interrupter control. The operation of this control causes the headlights to flash on and off as an overtaking signal, or as a warning when approaching crossroads. This is a very useful device for use late at night when an audible signal may not be given.

Another useful development is a time-switch for trafficators. The indicator arms remain out for a fixed period of time, and then return automatically. A great deal of confusion and annoyance is caused on the roads because people forget to return their trafficators to neutral after making a turn, and because automatic return devices have not operated by reason of the small angle turned through by the vehicle on forked road junctions, etc. The delayed action timing device eliminates this trouble, and the only thing that can be said against it is that it might return the indicators too early during a wait to turn right, across a continuous stream of traffic. This can, of course, be remedied by operating the switch again.

The Pop-Out cigarette lighter is a popular accessory in American cars and may well become a standard fitment in higher-priced British cars. In the Trico version, a button is pressed, the element heats up, and the unit pops out ready to be transferred to the cigarette for lighting. Replacement in the dark is made easy by a small pilot light in the empty socket.

NEW LEYLAND "COMET" 90 RANGE

A Redesigned Series to Replace the Original Models

A new range of cruiser-weight goods vehicles has been introduced by Leyland Motors Ltd. for both the home and export markets. Known as the "Comet" 90, the range replaces the existing "Comet" models, which were first introduced just over three years ago. In outward appearance the 90 series, so called because of their 90 h.p. diesel engines, bear a general similarity to the earlier "Comet" models, but there the resemblance ends. The new range has been designed on a more robust scale for gross ratings up to 11 tons 12 cwt., when used as a solo machine, compared with the 9 tons 5 cwt. of the original "Comet" range, thus almost entering the heavy duty class. As a tractor for articulated outfits, it is suitable for ratings up to 17 tons 17 cwt., compared with 15 tons 10 cwt. for the previous vehicle.

The new design includes a 351 cu. in. diesel engine, a 9 in. deep frame, body lengths up to 18 ft., five-speed helical toothed gearbox, a heavy duty two-speed rear axle, and a hydraulic braking system which operates on 16 in. and 15¾ in. diameter drums for the front and rear brakes respectively.

Three basic models are included in the range. There are two general haulage chassis, one for an 18 ft. long body and the other for a body up to 16 ft. 6 in. A 10 ft. 5 in. wheelbase chassis is also available as a tractor or for use as a dump truck. All are semi-forward control models with either right or left-hand controls. This arrangement is considered as being the best compromise between body length, weight distribution, engine accessibility, visibility and cab comfort particularly in tropical conditions.

In temperate climates the solo gross weighting is 26,000 lb. and in tropical zones 24,000 lb. These are maximum ratings irrespective of the capacity of the tyres. All models can be fitted with 9·00×20 or 10·00×20 low pressure, or 36×8 high pressure tyres. The rear axle limitation is two-thirds of the gross rating. To withstand this loading the Eaton two-speed heavy duty 16,500 rear axle has been chosen as standard for the range.

SPECIFICATION

ENGINE. Six cylinder oil direct injection. Bore and stroke 3·96× 4·75 in., capacity 351 cu. in. (5¾ litres). Maximum output 90 b.h.p. at 2,200 r.p.m. (governed). Maximum torque 240 lb. ft. at 1,300 r.p.m. Overhead valves. Seven bearing crankshaft.

CLUTCH. Single dry plate 13 in. diameter.

GEARBOX. In unit with engine and clutch. Five forward speeds and one reverse. Ratios 1: 1, 1·55: 1, 2·49: 1, 4·21: 1, 7·12: 1, reverse 7·45: 1.

REAR AXLE. Eaton 16,500 two-speed. Spiral bevel drive for high speed ratio, with epicyclic gear train reduction for low speed ratio. Standard ratios: high 4·89, low 6·80; alternative ratios: high 5·62, low 7·81, or high 6·14, low 8·52.

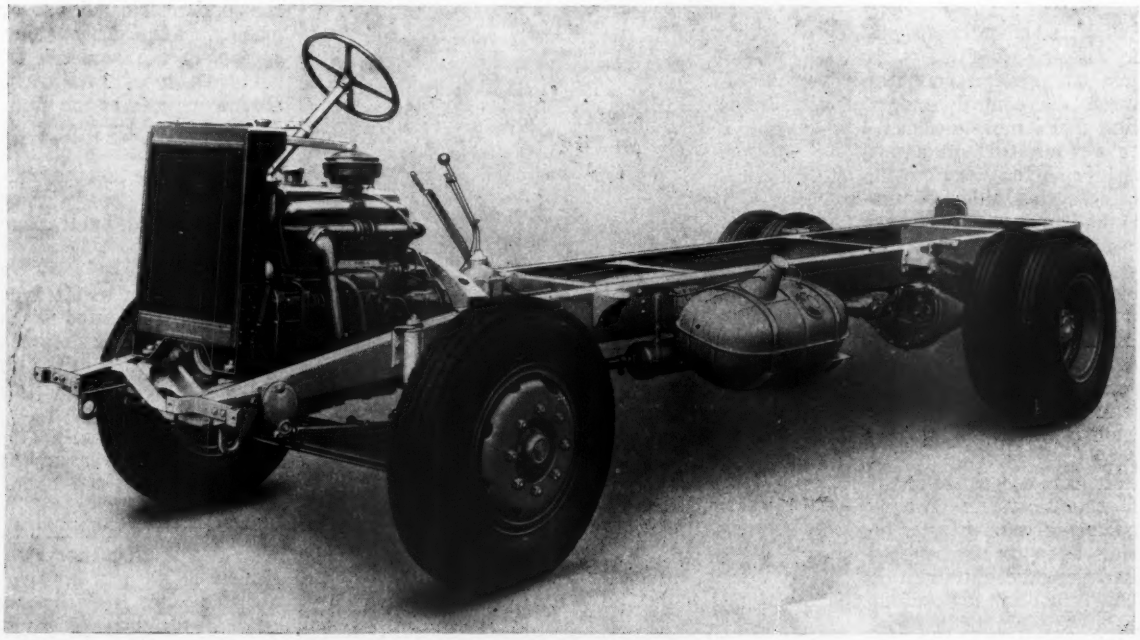
SUSPENSION. Semi-elliptic front and rear.

STEERING. Marles cam and double roller. Turning circles: 57 ft. on ECO2.1R and 1L, 43 ft. on ECO2.2R and 2L, and 60 ft. on ECO2.4R and 4L models when equipped with 9·00×20 tyres.

BRAKES. Two leading shoe internal expanding vacuum assisted on all four wheels.

Engine

Although following closely the very successful design of the older "Comet" diesel engine, the new unit is of greater capacity. It is a six-cylinder direct injection unit of 3·96 in. by 4·75 in. bore and stroke, giving a capacity of 351 cu. in. (5¾ litres). It develops 90 b.h.p. at its maximum governed speed of 2,200 r.p.m. and gives a maximum torque of 240 lb. ft. at 1,300 r.p.m. Its minimum full load fuel consumption is 0·362 pints per b.h.p. per hour, which compares more than favourably with 0·385 pints per



Leyland "Comet" 90 Chassis

b.h.p. per hour of its 300 cu. in. predecessor.

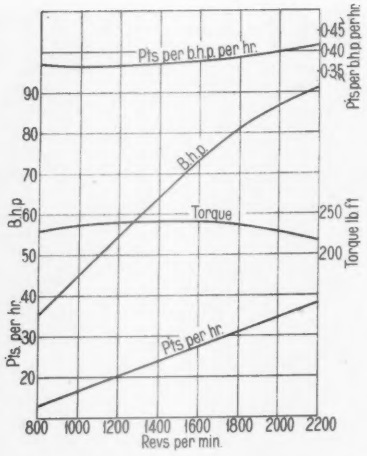
The engine, clutch and gearbox are of unit construction, and are carried in the frame by two rubber insulated mountings at the front and by two Leyland flexible link mountings at the rear. This combination of mountings, whilst providing stability, constrains the power unit to oscillate about its centre of gravity, thus reducing the transference of vibration to the chassis frame.

Of cast iron, the cylinder head is secured to the cylinder block by $\frac{11}{16}$ in. diameter high tensile steel studs having rolled threads. Increased swirl in the combustion chambers is produced by a lip-like formation on the interior surface of the valve portings. The exhaust valve seats are of "Valmet" metal and are frozen into machined recesses. The heat-treated silicon chromium steel valves are located in line in the cylinder head and are operated by push rods and rocker levers. Decompressor gear is fitted, operating on the exhaust valves. The crankcase, cylinder block and connecting rods are the same as in the previous range, apart from dimensional changes.

The seven crankshaft journals are 3·1 in. diameter and the crankpins 2·4 in. diameter, the latter being drilled eccentrically to reduce centrifugal loading and to act as sludge traps. Drilled oil-ways are provided for pressure lubrication to all bearings. A torsional vibration damper of the rubber bonded type is fitted externally to the front end of the crankshaft as before.

Of special aluminium alloy, the pistons are fitted with three compression and two slotted oil control scraper rings. The combustion chamber is of toroidal form and is machined in the piston crown. Fully floating gudgeon pins 1·3 in. diameter are used, and are retained in piston bosses by circlips. The gudgeon pins are offset towards the thrust side of the piston and this arrangement, in conjunction with the use of a special low expansion alloy mentioned above for the pistons, results in quieter running.

There is no change in the lubrication system, cooling system, or timing arrangements, but extra care has been taken with fuel filtration. A C.A.V. fuel injection pump is used in conjunction with Leyland Aphonic multi-hole injectors. This type of injector



Engine performance curves.

is a low pressure unit which subdues detonation by controlling flame propagation. It gives quieter running without in any way sacrificing reliability, power or economy. At the rear end of the injection pump is fitted a sensitive governor operated by vacuum through a large diaphragm. It gives full control of engine speed from stable idling at approximately 350 r.p.m. to maximum speed.

A diaphragm lift pump incorporating a hand priming device supplies fuel to the injection pump at constant pressure. It is mounted on the side of the injection pump and is operated mechanically by the camshaft of the main pump. A large capacity two-stage filter is introduced between the lift pump and the main pump, and additional filter elements are provided by Leyland designed edge filters which are fitted in each inlet adapter of the remote seat-type of injectors. Fuel leak-off from the injectors is returned direct to the main tank.

Front suspension and steering.

Transmission

The transmission is very nearly the same as in the previous range, except for the two-speed rear axle. This is of the Eaton 16,500 design and has a spiral bevel drive for the high-speed ratio with an epicyclic gear train reduction for the low speed ratio. These speeds are selected by a vacuum cylinder which is mounted on the axle casing, and which is controlled by a trigger mechanism that is located on the main gear change lever. The axle casing is of fabricated construction and consists of two steel pressings and hub end forgings which are flash welded to the casing. Three alternative axles, all of the two-speed type are available. These are the standard axle with a high-speed ratio of 4·89 : 1 and a low speed ratio of 6·80 : 1, an alternative axle with a high-speed ratio of 5·62 : 1 and low speed ratio of 7·81 : 1, and another alternative axle with ratios of 6·14 : 1 and 8·52 : 1 for the high and low speed ratios respectively.

Suspension

An outstanding feature of the design, the springing gives an exceptionally easy ride under all conditions of loading. Unusually long springs of the semi-elliptic, laminated type, 3 in. wide, are used, those at the front being 52 in. long with a maximum deflection of 3·75 in., and those at the rear being 54 in. long with a deflection of 3·9 in. incorporating helper springs. The shock absorbers of Newton and Bennett hydraulic type 8280 Mark 8 are fitted to the front axle and control the suspension efficiently on bump and rebound.

Brakes

All the brakes are of the two leading shoe design, and are operated hydraulically with vacuum servo assistance. The two leading shoe mechanism of the rear brakes is actuated when the vehicle is moving forwards or backwards. This is a great asset when reversing downhill.

The vacuum reservoir is of sufficient size to allow twelve consecutive brake applications. Front brakes are operated by dual internal cylinders on 16 in. diameter drums and at the rear the brakes are wedge operated on 15¼ in. diameter drums. The total effective liner area for the foot brake is 480 sq. in. and for the handbrake 294 sq. in.

Two-speed rear axle.

Frame

Of channel section, the frame side members are carbon steel pressings, swept down at the front end where they are tied by a robust shallow U-section front cross member. A length of reversed channel section is inserted in the frame side members to spread the load where the arched rear engine support member is located on the frame. The main cross members are of channel section riveted to the side members and additionally strengthened at their points of attachment by gusset plates. The frame stresses have been reduced by linking up wherever possible the road spring brackets with the cross members. An internationally accepted dimension of 34 in. has been adopted for the overall width of the frame. The substantial front bumper is carried on two brackets which extend from the front of the frame side members. Maximum frame section for all models is 9 in. by 3 in. by ¼ in. The haulage models have six cross members, whilst the tipper tractor models have four cross members.

Cab

Reinforced by carefully positioned pressings the super-structure of the cab has flexible rubber mountings at four points at the rear, with a single point front mounting carried on the centreline of the frame front cross member, so absorbing vibration and frame flexure. Access to the power unit is through a short rear-hinged bonnet and cowl in the cab, well insulated against heat transference. Provision has been made for easy removal of the engine. When a lever in the cab is operated, hot air from the engine is directed through a duct on to the interior of the windscreen.

Miscellaneous

The fuel tank is of welded construction and has a capacity of 24 gallons. It has a captive type filler cap and a magnetic gauge. It is mounted on the left-hand frame side member.

Detachable steel disc wheels with eight ⅞ in. diameter stud fixings are used, single on the front and twin on the rear. The wheels have sufficient offset to provide ample clearance between the tyres and springs for the use of snow chains, if desired.

24-volt lighting and starting is employed. The compensated voltage dynamo is of C.A.V. manufacture type G 524–4. The C.A.V. axial starter motor is type BS/524 K46. Two C.A.V. 12-volt batteries are utilised with a total capacity of 78 ampere hours.

Twin wind-toned electric horns, head and side lamps, stop and tail lamps are also supplied. The switch gear, fuses, cut-out and voltage regulator are combined in the instrument board which is also equipped with a speedometer, oil pressure and vacuum gauges and an ammeter. All instrument dials are indirectly lighted.

Full advantage has been taken of the Motor Industry Research Association's proving ground for the development of these new models. A prototype has been operating on the track for over six months and has already covered nearly 3,000 miles on the appalling half mile stretch of notorious Belgian pavé. This prototype has also travelled over 3,000 miles on the proving ground's cross-country rough track. The machine has stood up exceptionally well to these gruelling conditions and is still operating with its original set of road springs.

COMPRESSED AIR STANDARDS

THE first edition of *Compressed Air Terms and Standards* was published in 1932 by the British Compressed Air Society as an official reference for the compressed air industry, to establish definite standards for manufacturers and users of air and gas compressors, exhausters, rock drills, pneumatic tools, paint, and other spraying plant and appliances. A second edition was published in 1941, and because of the increase in demand, a third edition has now become necessary. In the formulation of this publication the British Compressed Air Society has collaborated with the British Standards Institution and other interested, and authoritative organizations.

The third edition has been considerably expanded so as to include additional British Standard Specifications, and more data on the installation, care and maintenance of pneumatic plant. This edition also elaborates the matter contained in the Society's *Handbook of Pneumatic Equipment*, and if used in conjunction with it will be found to deal with most technical aspects of the compressed air industry. The publishers claim that this is the only work which contains, in addition to the main British Standards relating to compressors and pneumatic tools, considerable information on the installation of pneumatic plant, together with tables and graphs. It is felt that the book will be of considerable use to manufacturers, and to users of compressed air.

Copies may be obtained, price 10s. 6d. each, post free, from the British Compressed Air Society, 94–98, Petty France, London, S.W.1.

INDEX AND BINDING CASES

The "Automobile Engineer" Index for 1950 is now on sale and copies are available, price 6d., by post 8d., or together with binding case 6s. 0d., 6s. 4d. by post.

Geo. W. King Ltd.

IN *The Automobile Engineer* for January 1951, a description was given of the materials handling methods employed in the new Vauxhall factory for Bedford commercial vehicles and references were made to power-pulled conveyors. The term "power-pulled" in connection with conveyors is a trade name used exclusively by Geo. W. King, Ltd., Hartford Works, Hitchin, Hertfordshire.

BONDING BRAKE LININGS

A Development of Interest to the Automobile Industry

By R. A. Johnson

A FURTHER stage in the development of adhesives for metals is marked by the use of synthetic resins for attaching linings to brake shoes and clutch plates. Considerable practical experience has now been accumulated with the adhesives used for this purpose and the methods of applying them in production. The only metal-to-metal adhesive that is approved by the Ministry of Supply for the bonding of load bearing aircraft structures is "Redux", manufactured by Aero Research, Ltd., Duxford, Cambridge. From this, a new material known as "Redux 64" has been developed. It has been specifically developed for bonding friction linings to metal.

The chief advantage of this material for its specific purpose is that it is formulated to give high strength at elevated temperatures, that is, temperatures in the region of 200-250°C. Usually resins for bonding metals have high strength at normal temperatures but begin to lose strength at temperatures above 70-80°C. For bonding friction linings it is not necessary to have quite such a high strength at normal temperatures because the lining is usually weaker than the adhesive. It is thus possible to forfeit some of the normal temperature strength in order to increase the hot strength. Table I shows the results of a series of tests to compare the strength obtained at temperatures of 20°C. and 200°C. using "Redux" and "Redux 64" as the adhesives and Woven Brake Linings of the same type and origin. From this it is seen that, although the strength of "Redux 64" at 20°C. is only 83% of that of "Redux", it is 53% greater at 200°C.

Table I

Comparison between strength of "Redux" and "Redux 64" at 20°C. and 200°C. with similar linings.

	Strength in P.S.I.	
	At 20°C.	At 200°C.
"Redux" ..	1575	447
"Redux 64" ..	1300	685

Advantages of Bonding

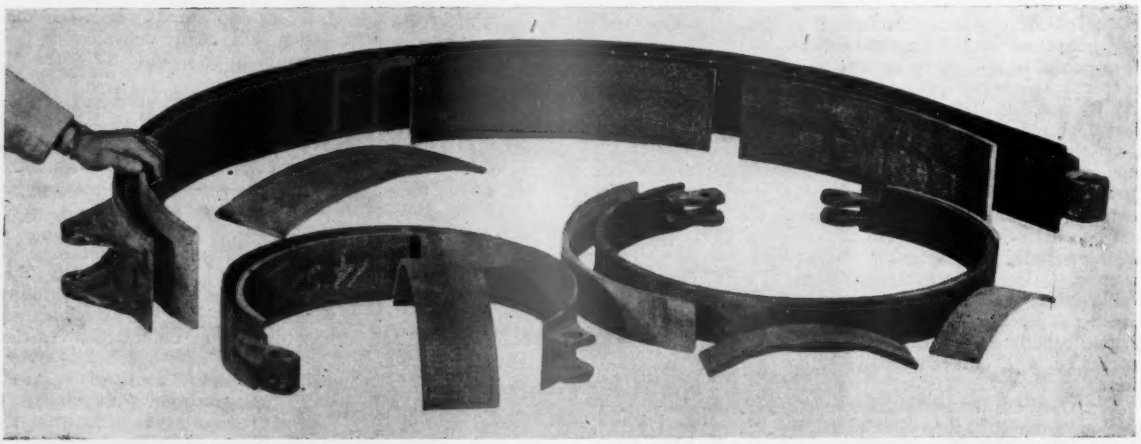
The use of adhesives for attaching brake and clutch linings offers several advantages over attachment by rivets although some claims for improved results have not met with unanimous acceptance. These advantages may be summarised as follows :—

(1) The life of the lining is longer, as the depth of the brake lining which is usable is not restricted by rivet heads but usually by the amount of adjustment on the brakes. It has been suggested that by suitable rivet disposition a riveted lining can also be worn down to the shoes, but this depends upon the design of the brake shoe and also upon the position of the pivot of the brake shoe; in order to have comparable life the riveted shoe must operate in such a manner that wear takes place from the centre of the lining.

(2) There is a greater surface area available for braking owing to the absence of rivet holes. This may account for as much as 7-10% of the area of the liner. This added friction area must increase the efficiency of the brakes if other conditions remain constant.

(3) The condition of the drum which after long use is free of score marks. These are usually caused by abrasive dust collecting above the rivet heads and by the rivet heads themselves when the linings have worn. It is sometimes argued that this cause of drum score is a popular fallacy and that as dirt and dust must go somewhere it is better that it should collect in the rivet holes than that it should be spread about on the surface of the liner. It has been proved, however, by the experience of many hundreds of thousands of miles running, that drums are preserved in excellent condition when bonded linings are used but that they are invariably badly scored when riveted linings are employed.

(4) There is less vibration or chatter in the lining because the lining is in intimate contact over the whole of its area. This is particularly true of large or heavily loaded clutch plates. In *The Automobile Engineer* for *September, 1945*, a description was given on the bonding of linings to clutch plates for military tanks. The life of the lining when bonded was approximately ten times greater than the riveted



A representative selection of brake bands for bonding. They are used for drag line equipment and excavators.

assembly because the whole area was in contact, whereas, when riveted, the lining moved in relation to the plate. This caused the rivet holes in the lining to elongate until such time as the lining was able to come away altogether. It is also found that there is a special need to bond high revving clutch plates as the linings tend to disintegrate between the rivets above certain speeds. This tendency is not present when a bonded assembly is used, since failure will usually occur in the metal before the adhesive or the lining. Because of this tendency the Borg and Beck clutch plates, as fitted to the Jaguar works-entered sports cars which were so successful in sporting events during 1950, were all bonded with "Redux".

(5) The heat transfer from lining to shoe is improved due to the completeness of the contact. This is important because, due to the many factors which are acting against the efficiency of the brakes in modern cars, e.g., increased speeds, smaller wheel sizes and body lines which shroud the wheels, the cooling of brakes is becoming a much greater problem.

(6) The attachment of the liner to the shoe is much stronger; this may be many times greater than the strength of a riveted assembly even at 250°C. It has been said that there is no necessity to increase the strength over and above that obtained by riveting. It is nevertheless satisfying to the user of the vehicle to know that the strength of attachment of the brake lining is in no way reduced but increased in order to obtain the other advantages of bonding.

(7) It is generally accepted that where bonding is carried out on a mass production scale the cost can be less than riveting. Although bonding requires the added time needed for curing the adhesive, this is not labour time and it is not necessary to drill and counterbore the rivet holes in the brake shoe and liners.

One of the main difficulties to be overcome is the guarded attitude of the engineer towards a process which is new. It is quite understandable that no one is going to recommend using a material or a process until it has been well proven. Adhesives have only been in use for bonding metals for about eight years, but during this time a considerable amount of experience has been accumulated. The de Havilland Aircraft Company, for example, have built over 300 passenger aeroplanes in which the skin structure of the wings and fuselage were bonded together with "Redux". These aircraft have now flown many millions of miles in all parts of the world, and at a recent conference of airline operators in New Jersey, U.S.A., a de Havilland representative stated that his company had not yet had a single demand for spares through the failure of the "Redux" method of bonding, although these aircraft have been operated at temperatures as low as –60°F. As a result the process has been used extensively in the de Havilland "Comet".

Table II
Showing difference in strength between linings made by different manufacturers.

	Failing Strength in P.S.I. at 200°C	Type of Failure
Type A	489	Lining
Type B	460	Lining
Type C	60	Adhesive
Type D	230	Lining
Type E	638	Lining
Type F	749	Adhesive

Of the items listed above, undoubtedly the longer life and the absence of drum scoring are the most important of the advantages offered. The Chief Engineer of a bus company reporting on his experience of bonded linings on double-deck local service buses, says "No difficulty has been experienced with any bonded lining . . . . The condition of the brake drums is excellent due to the absence of rivet holes, and the only criticism we have to make at this stage is the increased difficulty of removing worn liners".

Method of Bonding

Most engineers are familiar with the process of riveting liners, but few have yet had experience of bonding processes. It is therefore desirable to give an account of the process in order that comparisons may be made. The first and most important operation is that of degreasing the shoe and liners. This must be carried out by means of a trichlorethylene vapour degreasing tank, carbon tetrachloride or any satisfactory proprietary degreasing agent. After degreasing, it is important that the bonding faces shall not be handled as the film of grease which can be deposited by the fingers is sufficient to prevent good adhesion. After degreasing, the face of the brake shoe should be cleaned by shot blasting, pickling or sanding with a linishing belt or emery cloth. Any foreign matter on the lining, such as paint or printing ink should also be removed. If a liquid adhesive is being used, this should then be applied to one or both of the faces, according to the maker's instructions and allowed to stand for a period in order to allow solvents to escape or alternatively stoved at about 80°C. for a short time. The parts may then be clamped together and heated either by means of an oven or an electrically heated anvil. Sometimes the shoe is preheated before the liner, which has been previously coated with adhesive, is brought into contact. Dry forms of adhesive are available, in which case, after the cleaning process, the shoe and liner are brought together with a strip of adhesive film between them prior to clamping and heating. The heating cycle which is necessary for curing the adhesive may vary from three minutes at 400°C. to one hour at 145°C. depending upon the adhesive used, the form of heating employed, and the size of shoe being bonded. The larger the shoe, the longer the soaking time required.

Inspection of a bonded component is always a difficult matter when reckoned in terms of normal inspection technique. It usually means that control of the quality of workmanship takes on a more important aspect and the role of the inspector is that of a floating inspector who keeps a very careful watch on the materials used, method of applying the adhesive and the time and temperature cycle necessary for curing. It is also customary to carry out a destruction test either at regular intervals or as a spot check on an assembly selected by the inspector.

Types of Lining

Since the bonding of tank clutches started in 1944, many thousands of brake and clutch assemblies have been made. Amongst the items which are now being bonded daily are brake shoes for cars, lorries and buses, clutch plates for military vehicles, high performance sports cars, tractors and earth removing machinery, also brake bands for drag line equipment of up to 9 tons grab and brand bands for tractors, also the brakes on trawl winches. From this analysis it will be seen that the greatest volume of bonding is being done with the larger components. Some of the drag line equipment employs brake bands of up to 5 ft. diameter on which the cost of the lining is approximately £10 per band. The life of these is normally

increased by at least 15% resulting in an equivalent saving of 30/- or more per band.

It has been found that there is a considerable variation in the shear strength of various types of brake linings and that there is also a variation in the bond strength when using the same adhesive in conjunction with different types of linings. From this it would seem that there is a possibility of the constituents of the lining having some effect on the performance of the adhesive. It is therefore essential that every type of lining should be tested in conjunction with an adhesive before production bonding with these materials is carried out. Table II shows the variation between the shear strength obtained using linings of different origin but the same adhesive. It will be noted that in the majority of cases the lining itself fails before the adhesive, but there is one instance (Type C) where the lining materials affected the adhesive causing almost complete failure of the adhesive.

The Future of Bonded Linings

From present signs, it would appear that most re-lining will be carried out by Service Depots, or larger brake re-lining specialists and not by the smaller garage unit. There are several reasons why this should be so. First, in order to be economical a minimum turnover in excess of that likely to be handled by smaller garages, is necessary. Secondly, the processes involved demand a degree of cleanliness that is not normally obtainable in small shops. Contamination by oil or grease is inimical to good adhesion and successful results are not likely to be forthcoming if the same operative is required to drain oil sumps and bond brake linings. Moreover, a certain amount of equipment is necessary and the smaller unit might not find it possible to provide the necessary outlay for this kind of work. In some parts of the country a replacement service already exists for distributing bonded replacement shoes to garages daily so that the new shoes are on hand before the old ones are removed from the vehicle. The shoes taken from the vehicle are then collected the following day and returned to the re-lining depot for processing.

WOVEN PLASTICS

MESSRS. FOTHERGILL AND HARVEY, Ltd., have opened their new London Showrooms at 103, Mount Street, W.1, where an interesting plastic cloth for both standard trimming purposes and loose covers is on display. There are many very attractive features of this material, which is called "Tygan", and it merits serious consideration for trimming purposes. The manufacturers' claims for this material are as follows:

The pigment is actually mixed in with the plastic material of which the threads are composed. It is obvious, therefore, that the fade resistance will be greater than is the case with a dyed material, where the pigment is only on the surface of the threads. The plastic material will not dissolve or decay under the influence of any of the known atmospheric conditions in the world, and spilled oil, petrol, tea, etc., will not stain it or affect it adversely in any way. It can be cleaned easily with soap and water or any other cleaning fluid, as the material is not absorbent and does not shrink. It is woven from threads of plastic, and therefore air can pass through it and give adequate ventilation to the padding and interior of the seats.

It is claimed that Tygan wears as well, or better than leather cloth and does not cause clothes to become shiny. The plastic of which it is made will not support mould growth even in favourable tropical conditions, and it is not attacked by moths, ants or vermin of any kind.

In view of the insolubility of this material, it is suitable for use with almost any adhesive provided it dries quickly enough to prevent seepage of the liquid through the cloth. The adhesives recommended are, Bostic, rubber solution, Arabol and B.F. 76, made by the Arabol Manufacturing Co., of 8, Sans Wharf, London, E.C.1.

The most desirable feature of this material is undoubtedly the very attractive range of colours and patterns available. It certainly produces a clean, bright and very pleasing effect in a car interior. The manufacturers claim that the prices at which this material is marketed are competitive with those of other trim materials used for the same purpose. This may well be a very attractive substitute for other materials, particularly in view of the rising costs of some of them, especially wool. (1958)

Mobile Exhibition

TUFNOL, Ltd., of Perry Barr, Birmingham, 22B, announce that their new mobile exhibition began its tour on May 1 last. Although Tufnol is used extensively for a wide variety of engineering components, it is felt that the full possibilities of this modern material have yet to be explored. To give the greatest possible help to the rearmament and export drives in the country, therefore, it is proposed to use the mobile exhibition for Tufnol during the next twelve months, commencing with a tour of the Birmingham and district area.

The vehicle itself is of interest. It was designed within the Ellison organization of which Tufnol, Ltd., are an associated company. Internal lighting is provided entirely by fluorescent lamps, and an even temperature is ensured by a thermostatically-controlled heater, with adequate ventilation effected by small exhaust fans in the roof.

The driver's cab is specially designed to reduce fatigue on long journeys. The cab fittings include an adjustable armchair-type seat, an opening windscreen, and a mirror which can be adjusted by the driver so that he can view all parts of the interior exhibition to ascertain that the exhibits are riding well. The body was built on an A.E.C. chassis by Nudd Brothers and Lockyer Ltd., Kegworth, Derby. (1960)

Car Engine Heater

MESSRS. GEO. BRAY and Co. Ltd., of Leicester Place, Leeds, 2, had on their stand at the Castle Bromwich Section of the British Industries Fair a range of car engine heaters for use in frosty weather, and for vehicles such as fire engines, ambulances, and some commercial vehicles where a quick start in cold weather is essential. Some car manufacturers make special provision for fitting these units, but in other cases they may be inserted in the radiator hose connection, and for this purpose there are several designs to suit almost all popular makes of car. It is simply an electrical heater element into which a lead can be plugged from an external main supply of electricity. An extension to the dash can be provided at extra cost, so that it is unnecessary to open the bonnet and plug in or disconnect the mains lead.

Two sizes of unit are offered for sale for 200 to 250 volt mains. A 150-watt version is provided for engines of up to 16 h.p. (R.A.C. rating) and a 250-watt unit for engines of over 16 h.p. Both are offered at reasonable prices.

It is claimed that reduction of engine wear is obtained, since a considerable proportion of the total wear takes place when warming and starting up. The life of the battery and starter motor is prolonged, and fuel economy obtained by warm starting without excessive use of the choke. These economies alone might well save the total cost of the heater in a year or two, and the advantages of easy starting, and no freezing worries in cold weather make this a very useful accessory. (1954)

Institution of Mechanical Engineers, Automobile Division

VALVE GEAR DESIGN

An Analytical Approach to Automobile Engine Requirements

By J. L. H. Bishop, A.M.I.Mech.E.*

In the automotive field during the last decade, owing primarily to increases in maximum engine speed and the almost universal use of overhead valve gear, increasing attention has been focused upon improper valve functioning which causes noisy operation, excessive valve spring surge, valve seat pounding, and exhaust valve burning.

With the conventional design of cams no allowance is made for the flexibility of the push rod and rocker, consequently in general the accelerations imparted to the valve are much greater than is necessary at high speeds when inertia effects build up.

A general mathematical analysis of the conventional three-arc cam is presented, introducing the terms "eccentricity" and "instantaneous radius", and this is followed by a similar analysis of the new cam profile using the same terms, together with the method of correcting the cam profile for high-speed operation. It is also shown how the new general equations can be used to deduce a variable eccentricity ramp, the purpose of which is to take up clearance at variable velocity.

The effects of the new cam profile on horse-power and thermal efficiency are shown; the effect of the acceleration characteristic on valve spring design also is illustrated, and a comparison shows the reduction in spring weight made possible by the elimination of valve spring surge.

Introduction

As the push-rod-operated valve gear is now used almost universally for the popular high-performance car, the new cam contour has been primarily designed for this type of operation. The design of a cam contour is governed by the type of follower employed, and by the desired lift, velocity, and acceleration to be imparted to the valve; moreover, the maximum value of the negative acceleration and the characteristic shape of the acceleration diagram at the peak engine-speed influence spring design, since the magnitude and shape of the diagram determine the maximum load and rate of the spring to be used. Cam followers may be of either a flat-faced or a roller type. Flat-faced followers are used extensively for automotive engines operating at high speeds, owing to their possession of certain inherent advantages over the roller type of follower. They are lighter, of more economical construction, and more easily lubricated.

General practice

The three-arc type or so-called harmonic cam shown in Fig. 1 is in general use. This type has been used for many years and consists of a nose radius and two flank radii connecting the nose radius to the base circle. The formulae for designing a cam of this type are well-known and can be obtained from any suitable text-book (Heldt)†. For taking up clearance it is usual to join the base circle to the clearance circle by a spiral or what is generally known as the constant velocity ramp.

The three-arc cam

The principles of the three-arc cam are here presented in a new manner (see Fig. 2) For convenience, since the three-arc cam is symmetrical, halves only of the lift, eccentricity, and instantaneous radius curves are shown in Fig. 3, which depicts a mathematical analysis of a three-arc cam.

Between $x=0$ and $x=x_1$ the lift curve is

$$y=R_1(1-\cos x) \quad . \quad . \quad . \quad (1)$$

the eccentricity curve is

$$\frac{dy}{dx}=R_1 \sin x \quad . \quad . \quad . \quad . \quad . \quad (2)$$

and the instantaneous radius curve is

$$\frac{d^2y}{dx^2}=R_1 \cos x \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where $R_1=r_1-r_0$, x=angular travel of camshaft in radians, eccentricity=velocity of follower/ω, instantaneous radius=acceleration of follower/ω^2, ω=angular velocity of camshaft in radians per second.

Between $x=x_1$ and $x=x_2$, the lift curve is

$$y=R_2 \sin (x+a)-b \quad . \quad . \quad (4)$$

the eccentricity curve is

$$\frac{dy}{dx}=R_2 \cos (x+a) \quad . \quad . \quad . \quad (5)$$

and the instantaneous radius curve is

$$\frac{d^2y}{dx^2}=-R_2 \sin (x+a) \quad . \quad . \quad (6)$$

where $R_2=D$, and a and b are constants.

Referring to the instantaneous radius curve, since the acceleration is directly proportional to the instantaneous radius of the cam, the acceleration theoretically makes an instantaneous change at the beginning and end of the lift curve, and at the points of maximum eccentricity (or velocity). At the start of the lift curve the inertia load is supposedly applied instantaneously and, at the points of maximum velocity, the positive inertia load disappears while a negative inertia load appears instantaneously. The negative inertia load is resisted by the action of the valve spring, and such rapid change in load at these points can be likened to impact loading. With a valve gear of finite rigidity, this instantaneous change in load cannot take place as the components deflect and part company, the valve and valve spring acquiring a new motion at every increase of a few hundred revolutions per minute. It is this modification of the valve motion which constitutes valve-spring surge.

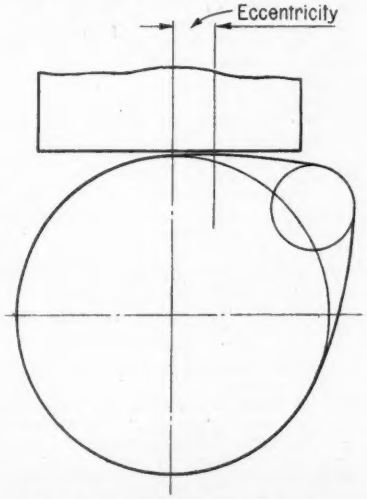


Fig. 1. Harmonic cam and follower.

Neglecting resonance of the harmonics of the lift curve, and the natural frequency of the valve spring (Wahl) which sometimes results in increasing surge amplitudes to $\frac{3}{16}$ inch or more, for the non-mathematically minded the following simplified explanation may make the phenomena of surging seem less mysterious. In a badly designed cam the change in theoretical acceleration of a valve might be in the region of 600 times the acceleration due to gravity at maximum engine speed. Consider now the first effective coil of a valve spring suitable for a 16 h.p., four-cylinder engine; the weight of this portion of spring might easily be 0·03 lb. Upon being accelerated the inertia of this bottom coil will resist in the direction of motion, and when the acceleration suddenly reverses the one coil alone will exert a force of 18 lb., and assuming a spring rate of 180 lb. per in. this single coil, becoming instantaneously decompressed, will jump 0·1 inch and initiate a wave in the spring. In a push-rod-operated valve mechanism this effect is accentuated by the flexibility of the long push-rod. For example, referring to the first interval of the cam profile or flank which serves to accelerate the valve to maximum opening velocity, this in a 115 deg. camshaft period might have a designed interval of about 16 deg., but at maximum speed, owing to the shortening by compression of the push-rod, the interval might easily be reduced to 12 deg.; consequently the real acceleration of the valve will be considerably in excess of the designed value based on that of the cam profile. The effect of this shortening of the time during which the valve is being lifted to maximum velocity results in the valve attaining a velocity far in excess of the theoretical maximum, and the valve will commence an independent motion of its own until such time as the valve spring restores the equilibrium, the rocker meanwhile contacting the valve and the latter taking a further leap off the springboard. It is the constant repetition of a particular motion at any given speed that constitutes the phenomenon of valve-spring surge.

Principles of cam design

General. The motion of a cam-operated valve is depicted in Fig. 4 and can be described in four intervals. During the first interval the valve is lifted from its seat

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†An alphabetical list of references is given in the Appendix, p.10.

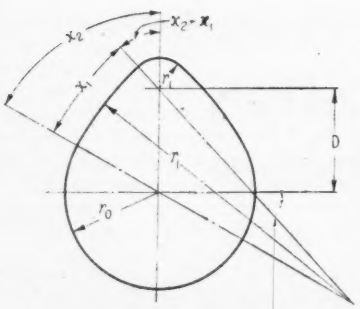


Fig. 2. Three-arc cam.

x_1 Angular measure of flank radius.
x_2 Angular measure of half the nominal cam period.
$x_2 - x_1$ Half the angular measure of the nose radius.
r_1 Flank radius.
r_2 Nose radius.
r_0 Base circle radius.
D Maximum negative instantaneous radius.

and accelerated to maximum velocity; during the second interval the valve is decelerated from maximum velocity to zero velocity at the top of the lift, the decelerating force being provided by the valve spring which yieldingly opposes the movement of the valve in its opening motion. During this latter interval the acceleration reaches its maximum negative value at the top of the lift.

At the beginning of the third interval the valve, now at zero velocity, commences on its return path under the accelerating force of the spring and reaches its maximum closing velocity; at this point, continuing into the fourth and final interval, the valve is decelerated by the cam and finally brought to rest.

In designing cams the mathematical expressions defining the lift curve or, alternatively, the radii determining the geometry of the cam profile often are, but need not necessarily be, symmetrical for the opening and closing side of the cam profile. In order to minimize the load on the valve spring and to reduce the distortion of the valve mechanism at high engine speeds it is usual to aim at a maximum positive acceleration of about twice the negative acceleration.

Requirements of the ideal Contour. The problem of designing a suitable contour therefore resolves itself into choosing a shape which induces the valve and its associated components to move in a specified manner, the motion chosen to be fundamentally unaltered, notwithstanding valve train flexibility, throughout the entire running range of the engine; the acceleration characteristic should be such that a spring of high frequency can be used, so that the chosen harmonics of the cam motion cannot be excited in the running range. Although there are numerous mathematical functions for theoretically determining a suitable lift curve, those which permit the necessary corrections for valve train flexibility are limited. The lift curve chosen for the new contour permits a simple mathematical study of the motion of the actual valve, taking into account the flexibility of the mechanism. It is therefore possible to put into the design of the contour the necessary corrections for reducing the accelerations.

General Principles of the New Contour. The lift curve finally chosen for the new patented design and giving a suitable opening diagram is made up of three basic functions or intervals in series as depicted in Fig. 5. The first function consists of a negative half-period sine-wave combined with an ascending straight line, the resultant curve commencing with zero acceleration, and harmonically accelerating the valve from zero velocity, or some predetermined velocity, to maximum opening velocity and zero acceleration; at this point it is followed by the second function consisting of a second half-period sine-wave commencing at zero acceleration and with the same maximum opening velocity as that attained at the end of the first function, this second half-period sine-wave decelerating the valve to zero velocity at maximum valve lift, and again accelerating the valve to maximum closing velocity and zero acceleration. The third function is the same as the first function but in reverse order, consisting of a negative half-period sine-wave combined with a descending straight line, and serves to decelerate the valve from maximum closing velocity to zero velocity or some predetermined velocity. The valve is returned to its seat with zero acceleration, thus minimizing the final compression and distortion of the components comprising the valve gear.

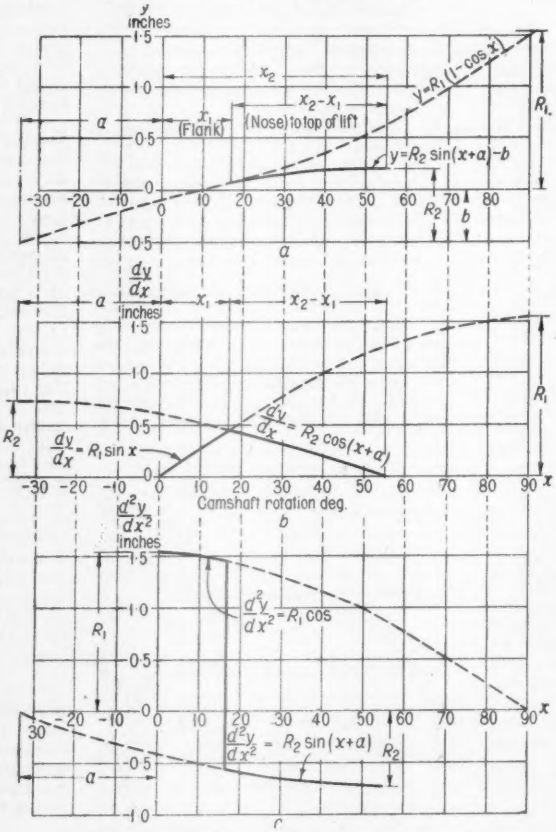


Fig. 3. Mathematical analysis of three-arc cam.

a Lift curve.
Note.—Zero lift is at 0 deg.
b Eccentricity curve.
Velocity of follower = $\omega \times$ eccentricity, where ω = speed of camshaft rotation in radians per second.
c Instantaneous radius curve.
Acceleration of follower = $\omega^2 \times$ instantaneous radius.

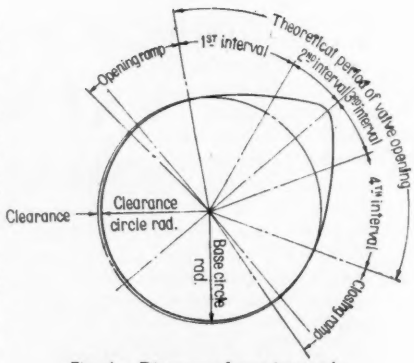


Fig. 4. Diagram of cam intervals.

The general expressions for defining the cam profile are as follows :—

Let m_1 = the number of periods per revolution of camshaft for the first interval,

m_2 = the number of periods per revolution of camshaft for the second and third intervals,

n = the constant slope added to the negative half-period sine-wave comprising the first interval,

R_1 = the amplitude of the half sine-wave of the first interval of the lift diagram,

R_2 = the amplitude of the half sine-wave comprising the second and third intervals,

x = angular movement of camshaft in radians, and

y = displacement or lift.

The general expressions defining the first interval are :—

for lift

$$y = -R_1 \sin m_1 x + nx \quad . \quad (7)$$

for eccentricity

$$\frac{dy}{dx} = -m_1 R_1 \cos m_1 x + n \quad (8)$$

for instantaneous radius

$$\frac{d^2y}{dx^2} = m_1^2 R_1 \sin m_1 x \quad . \quad . \quad (9)$$

The general expressions defining the second and third intervals, assuming for convenience a change in the axis to the commencement of the second interval, are :—

for lift

$$y = R_2 \sin m_2 x \quad . \quad . \quad . \quad (10)$$

for eccentricity

$$\frac{dy}{dx} = m_2 R_2 \cos m_2 x \quad . \quad . \quad (11)$$

for instantaneous radius

$$\frac{d^2y}{dx^2} = -m_2^2 R_2 \sin m_2 x \quad . \quad (12)$$

where eccentricity = velocity of follower/ω, instantaneous radius = acceleration/ω^2, and ω = speed of camshaft rotation in radians per second.

The geometrical relationship of eccentricity and instantaneous radius has been dealt with elsewhere (Candee).

Simple Contour without Ramps. For a particular application of a symmetrical multi-sine-wave cam in its simplest form, uncomplicated by the addition of ramps and so designed to commence with zero velocity, the conditions should be as follows: in the instantaneous radius curve, the amplitudes and periods of the sine-waves of the first and fourth intervals should be equal, the amplitude of the negative sine-wave comprising the second and

third intervals should be half that of the first interval, and the period of this second sine-wave four times that of the first interval; consequently the area above the axis for the first interval should be equal to the area below the axis of the second interval. If the instantaneous radius (or acceleration) curve were now integrated, the eccentricity (velocity) curve would result, the ordinates of the curve being the eccentricity of the cam. The first interval is represented by the half-period of a negative cosine-curve plus a constant; the combined curve commencing at zero velocity, the maximum velocity is equal to double the wave amplitude; the second and third intervals are now represented by the half-period of another cosine-wave, and its amplitude is double that of the cosine-wave comprising the first interval. The fourth interval is another cosine-wave plus a constant identical with that of the first interval. If the velocity curve were now integrated the lift-curve of the cam would be obtained; the ordinates of this curve are the vertical displacements of the follower.

Since the eccentricity is zero at 0 deg.,

$$n = m_1 R_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

and since eccentricities must be equal at the end of the first interval,

$$n + m_1 R_1 = m_2 R_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

$$m_1 = 4 m_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

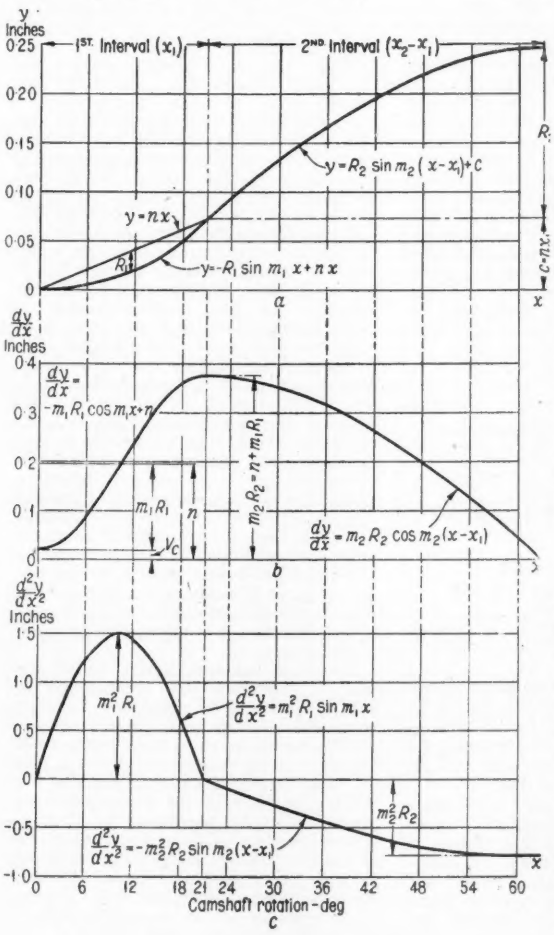


Fig. 5. Mathematical analysis of multi-sine-wave cam.

a Lift curve.
b Eccentricity curve.
Velocity of follower = $\omega \times$ eccentricity, where ω = speed of camshaft rotation in radians per second.
c Instantaneous radius curve.
Acceleration of follower = $\omega^2 \times$ instantaneous radius.

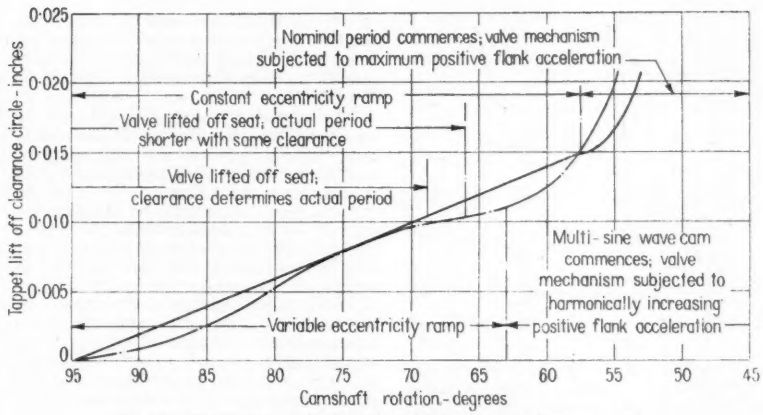

Tappet lift off clearance circle - inches
Nominal period commences; valve mechanism subjected to maximum positive flank acceleration
Constant eccentricity ramp
Valve lifted off seat; actual period shorter with same clearance
Valve lifted off seat; clearance determines actual period
Multi-sine wave cam commences; valve mechanism subjected to harmonically increasing positive flank acceleration
Variable eccentricity ramp
Camshaft rotation - degrees

Fig. 6. Comparison of ramps for 115 deg. nominal cam period.

Centre line of cam nose is at 0 camshaft degrees.
——— Constant eccentricity ramp.
— - — Variable eccentricity ramp.

Let λ = maximum value of negative acceleration of cam (or value of maximum negative instantaneous radius). This corresponds to dimension *D* on Fig. 2, and is the most important cam dimension;

l = maximum lift of cam.

The maximum negative value of instantaneous radius

$$\lambda = \frac{8 m_2^2 l}{\pi + 8} \quad . \quad . \quad (16)$$

and the maximum value of the positive instantaneous radius (proportional to the maximum flank acceleration) is double this value. The value of m_2 is determined from the period of the valve, hence for a given lift the maximum accelerations can be immediately calculated. From the other simple relationships the equations of the cam profile can be written down and values of lift and eccentricity calculated for each degree of camshaft rotation.

The Contour for Operation with Ramps. In the design of camshaft profiles for push-rod-operated valves it is usual to lead up to the flank of the cam by what is usually known as the ramp, a purpose of which is to take up tappet clearance between the clearance circle and the base circle of the cam. In an ordinary cam it is quite usual to arrange curves so that the first small part of the lift from the clearance circle takes the form of a constant eccentricity ramp on the opening side and a similar ramp on the closing side, the two slopes being, however, often made unequal. Theoretically, the slope of the ramp determines the velocity at which the valve is lifted off its seat and also the closing velocity of the valve; in actual practice, however, in the older type of cams with constant eccentricity ramps, at high speeds the valve unseats itself twice when both opening and closing. The reasons for this are obvious from what has already been said on valve train flexibility and spring surge. In addition, as engine speed increases, the actual opening period is progressively shortened, owing to the increasing delay in opening and to the earlier closing of the valve.

In designing a multi-sine-wave cam it is first necessary to calculate the amount of compression in the valve mechanism at a speed slightly in excess of that of the maximum speed of camshaft rotation. Since the change in the length of the push-rod will now change at the rate of the chosen sine-curves or harmonics of the motion, it is possible to reduce the amplitudes of the sine-waves chosen for determining the cam profile by the estimated maximum deflexion due to the inertia forces at the chosen speed. In this way the strain energy in the push-rod is gradually built up and again gradually returned to the moving system.

Generally speaking the designed opening period of a multi-sine-wave cam will be about 10 camshaft degrees longer than the required nominal period, owing to the fact that the valve is opening with zero acceleration harmonically increasing to its maximum value. In spite of the theoretical increase in period, in practice, taking clearance into account, the actual time during which the valve is off its seat at any given speed is slightly shorter than that of a corresponding three-arc cam with a constant eccentricity ramp.

Fig. 6 shows the lift diagram of the first portion of the opening side of the two cams. The closing side is symmetrical with that shown and has therefore been omitted.

In the three-arc cam the constant eccentricity ramp is joined up to the flank portion of the cam at 57 deg. 30 min. In the multi-sine-wave cam the variable eccentricity cam ends at 63 deg. The camshaft angle at which the valve is lifted and returned to its seat is determined by the amount of clearance, measured in the case of a side-valve engine at the tappet, and in an overhead-valve engine between the rocker and the valve stem. In the older type of cams it is the general practice for the ramp to end and the flank radius

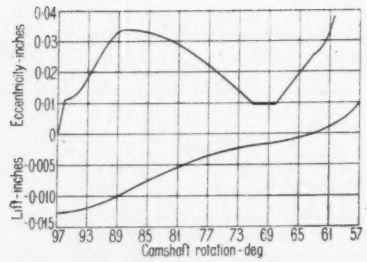


Fig. 7. Multi-sine-wave ramp for 115 deg. nominal cam period.

Centre line of cam nose is at 0 camshaft degrees.

to commence at 0·015 inch above the clearance circle, and the nominal period for timing purposes commences at that arbitrary point. In practice, however, the running clearance, if measured at the tappet, is always set at something smaller than 0·015 inch to ensure that the valve is opened and shut on the ramp portion of the cam. In general, therefore, the actual period of the valve opening will be longer than the nominal period and is determined by the running clearance. In Fig. 6, since the rocker ratio of the overhead valve gear is 1·42/1, and the running clearance measured at the valve, for convenience, is 0·015 inch, the valve will just be lifted off its seat when all the clearance has been taken up, that is, when the tappet has been lifted 0·015 inch divided by the rocker ratio, or 0·0105 inch approximately, as shown in the diagram.

The period of the cam establishes the number of complete sine-waves per revolution, that is, m_1 and m_2. Knowing m_1 and the amount of compression in the mechanism at the designed speed, since the compression is sinusoidal, the first differential of the compression wave establishes the variation in eccentricity between zero and the maximum designed speed. The eccentricity assumed at the commencement of the mathematical cam profile can now be determined and thus the amplitudes of the two sine-waves defining the lift-curve can be calculated. The positive and negative accelerations of the cam profile are generally about 85 per cent of the values determined on the basis of zero velocity, this being the assumption made when determining arc radii of the older type cam.

If V_c is the eccentricity at the commencement of the cam profile, then, from equation (8),

$$V_c = n - m_1 R_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

Since the eccentricities (or the slope of the lift curve) must be identical at the end of the first interval and the beginning of the second interval, then

$$m_2 R_2 = n + m_1 R_1 \quad . \quad . \quad . \quad . \quad . \quad (18)$$

If the periodicities of the sine-wave of the first interval and that of the second and third intervals remain in the ratio of four to one as in the idealized cam previously described, it will be found that the introduction of end velocity slightly reduces the two to one relationship between the maximum positive and maximum negative accelerations. Expressed mathematically the maximum negative instantaneous radius

$$\lambda = \frac{m_2(8m_2 l - \pi V_c)}{\pi + 8} \quad (19)$$

and the maximum positive instantaneous radius

$$m_1^2 R_1 = \frac{4m_2[4m_2 l - V_c(4+\pi)]}{\pi + 8} \quad (20)$$

The amplitude of the sine-wave of the first interval

$$R_1 = \frac{4m_2 l - V_c(4+\pi)}{4m_2(\pi+8)} \quad (21)$$

The numerical values can now be put in the equations, and the expressions defining the cam contour can be written down.

Using seven-figure mathematical tables and some form of calculating machine, the values of lift and eccentricity are calculated in camshaft degree intervals. The values are tabulated on the camshaft detail drawing together with the tangent height above the camshaft centre, and this provides the tool-maker with all the information necessary to make the master cam.

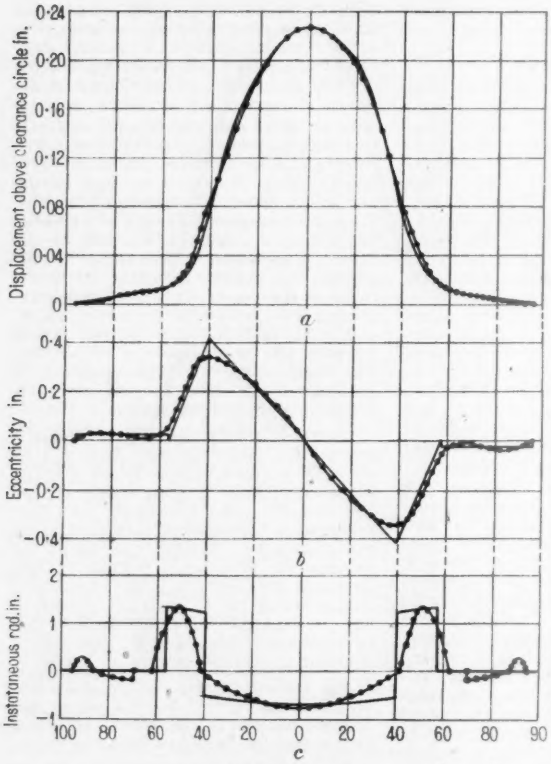


Fig. 9. Comparison of three-arc and multi-sine-wave cams

a Lift curve.
Lift, 0·214 inch } nominal.
Period, 115 deg. }
b Eccentricity curve.

Theoretical velocity $= \frac{\omega}{12}$ eccentricity in feet per second, where ω = speed of camshaft rotation in radians per second.

Note.—Reduction in eccentricity permitting lighter, smaller diameter tappet.

c Instantaneous radius curve.

Theoretical acceleration $= \frac{\omega^2 \times \text{instantaneous radius}}{12}$ ft. per sec.^{2}.

——— Three-arc cam.
—●—●— Multi-sine-wave cam.
Centre line of cam nose is at 0 camshaft degrees

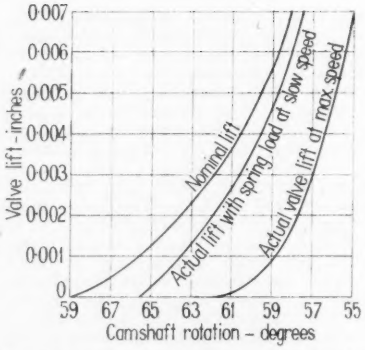


Fig. 8. Variations in opening conditions due to spring load and inertia.

Centre line of cam nose is at 0 camshaft degrees. Clearance 0·015 inch at valve 115 deg. period multi-sine-wave cam.
Nominal timing—0·022 inch clearance at valve.

For example, the equations defining a practical cam of 120 deg. nominal period and 0·235-inch lift are :—

between $x=0$ and $x=x_1$:—

$$y = -0{\cdot}01905 \sin 8\tfrac{1}{2}x + 0{\cdot}18528x,$$
$$\frac{dy}{dx} = 0{\cdot}15878 \cos 8\tfrac{1}{2}x + 0{\cdot}18528,$$
$$\frac{d^2y}{dx^2} = 1{\cdot}3232 \sin 8\tfrac{1}{2}x.$$

and between $x=x_1$ and $x=x_2$:—

$$y = 0{\cdot}16515 \sin 2\tfrac{1}{12}x,$$
$$\frac{dy}{dx} = 0{\cdot}34406 \cos 2\tfrac{1}{12}x,$$
$$\frac{d^2y}{dx^2} = -0{\cdot}71680 \sin 2\tfrac{1}{12}x.$$

One of the specific objects of the ramp is to take up lost motion without shock, but with the constant eccentricity type this is physically impossible except at low engine speeds. The slope of the opening and closing ramps influences the velocity at which the valve will be lifted and returned to its seat, the actual period in camshaft degrees during which the valve is actually off its seat depends on spring load and becomes shorter as engine speed increases owing to inertia effects (see Fig. 8), and the slope of the ramps at low speeds determines the actual period of the valve, taking into account the working clearance. Therefore, in a constant eccentricity ramp a smaller slope (or eccentricity) improves valve seating conditions, but for a constant working clearance it would have the disadvantage of increasing the actual opening period and this would result in burning of the exhaust valves owing to a too early opening.

The Multi-sine-wave Ramp. In the new patented multi-sine-wave ramp design the velocity at which the clearance is taken up is variable. The tappet is at first lifted very slowly, the clearance is then partly taken up at increasing velocity, the tappet motion is next slowed down gradually to the position at which all the clearance is taken up, and the valve is gradually lifted off its seat at a velocity of about one-third of that of the conventional ramp. The transition between the small slope of the ramp where the valve is lifted off its seat, and the slope or eccentricity at the commencement of the multi-sine-wave cam already determined mathematically for high-speed operation, is so arranged that the characteristic motion of the valve and valve spring is unaltered throughout the running range of the engine in spite of the fact that the relative opening velocity of the valve

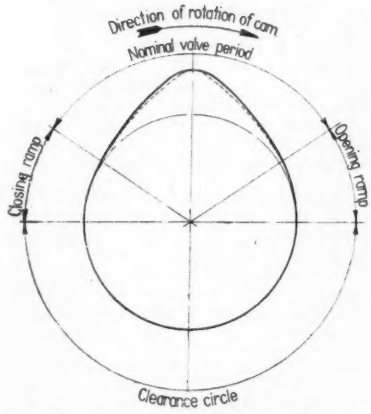


Fig. 10. Comparison between three-arc and multi-sine-wave cams.

———— Multi-sine-wave cam.
— — — —Three-arc cam.

decreases with increasing engine speed owing to inertia effects. The transition is effected by what might be termed "the use of a mathematical springboard for the valve take-off." For about 5 deg. of camshaft rotation a constant acceleration ramp joins the low eccentricity portion of the ramp to the higher eccentricity portion at the commencement of the mathematical cam (see Fig. 7).

At low speeds the valve is lifted at low velocity by the constant eccentricity portion of the ramp and its velocity is then accelerated during the 5 deg. motion of the transition ramp. At high speeds, due to inertia effects, the period shortens progressively until at the highest speed the valve does not move materially until the 5 deg. of transition are taken up in lost motion, and then the valve leaves its seat with a low velocity. The valve is therefore unseated and seated at low velocity irrespective of the speed of rotation of the camshaft.

The variations in opening conditions due to spring load and inertia are depicted in Fig. 8.

The mathematical expressions for designing the multi-sine-wave ramp are similar to those for the new cam profile.

In Fig. 9 is shown a comparison of theoretical lift, eccentricity, and instantaneous curves for the two types of cam of the same lift and nominal periods, complete with opening and closing ramps.

A comparison of the contours for the two types of cam of the same lift and period is depicted in Fig. 10. The actual nose radius of the multi-sine-wave cam is generally greater than that of the three-arc cam, thus reducing the maximum contact stresses.

The Cam in Practice. Owing to the decrease in acceleration forces of the new cam profile, it was found possible to increase the valve lift on one engine from 0·360 inch to 0·390 inch, with the same size camshaft bearings, and without increase in the maximum valve spring stresses. The comparison of the same engine tested with the two types of cam profile is shown in Fig. 12. The improvement in thermal efficiency is attributed to the improved valve motion and correct valve timing at all speeds throughout the running range of the engine.

Valve spring design

By raising the rate of a spring the spring surge is reduced, since the raised natural frequency moves some of the harmonic orders outside the running range of the engine. With the three-arc cam, owing to the characteristic shape of the negative acceleration, there is a practical limit for the spring rate since it increases the maximum stress of the spring. The characteristic shape of the multi-sine-wave cam is such that a suitable spring can have a high rate and frequency without a corresponding increase in maximum load and stress.

A comparison of valve spring acceleration relationships of springs working in conjunction with three-arc and multi-sine-wave cams is depicted in Fig. 11.

Owing to the large reduction in maximum spring surge amplitude with the multi-sine-wave cam there is also evidence to suggest that much higher nominal maximum stresses and stress ranges can be used than with a spring designed for a three-arc cam. For example, in the older type the nominal stresses were about 48,700–86,500 lb. per sq. in. in order to eliminate the possibility of breakage. On the other hand an experimental engine using a non-surge cam has completed more than 10,000,000 stress reversals at nominal stresses of 63,700–116,900 lb. per sq. in. The reason for this is that in the former spring, owing to the increase

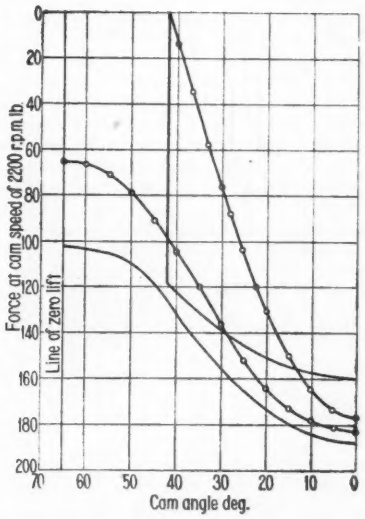


Fig. 11. Valve spring acceleration relationships.

———— Three-arc cam with 4½-coil spring—162 lb. per in.
—○—○— Multi-sine-wave cam with 3½-coil spring—208 lb. per in.
Centre line of cam nose is at 0 camshaft degrees. Rocker ratio 1·42/1.

in stress due to surge, the actual stress range for the bottom coil is 4,400–100,600 lb. per sq. in.

A comparison of the springs on a pound-age-weight basis shows in a more striking manner the advance which the design of the non-surge cam has made possible. In the case of the 4½-coil spring the ratio maximum poundage/weight of spring is 44 lb. per oz.; this ratio for the 3½-coil spring is 72 lb. per oz., an increase of 64 per cent; comparing the ratio nominal stress range/weight of spring, in the former the ratio is 12,600 lb. per sq. in. per oz., and in the latter 21,100 lb. per sq. in. per oz., an increase of 67 per cent.

Fig. 12 depicts the improvement in brake horse-power and thermal efficiency obtained by using a multi-sine-wave cam with a suitable spring instead of a three-arc cam with a suitable spring.

Summarizing the particulars given in Table I, in the case of the three-arc cam

TABLE I. COMPARISON OF VALVE SPRING SURGE CHARACTERISTICS.

Extracted from Development Report.

Engine, r.p.m.	Three-arc cam with 4½-coil springs—natural frequency 20,400 cycles per min.		Multi-sine-wave cam with 3½-coil springs—natural frequency, 26,200 cycles per min.	
	Degree of surge	Harmonic order	Degree of surge	Harmonic order
1,200–2,400	Slight	34th–17th	None	
2,500	⅛–¼ inch	16th	,,	
2,600	,, ,,	16th	,,	
2,700	,, ,,	15th	,,	
2,800	,, ,,	15th	,,	
2,900	,, ,,	14th	,,	
3,000	,, ,,	14th	,,	
3,100	,, ,,	13th	,,	
3,200	,, ,,	13th	,,	
3,300	,, ,,	12th	,,	
3,400	,, ,,	12th	,,	
3,500	,, ,,	12th	,,	
3,600	,, ,,	11th	,,	
3,700	,, ,,	11th	1/16–⅛ inch	14th
3,800	,, ,,	11th		14th
3,900	Strong coil clatter	10th	⅛ inch on	13th
4,000	,, ,, ,,	10th	most	13th
4,100	,, ,, ,,	10th		13th
4,200	Valve crash		Valve crash	

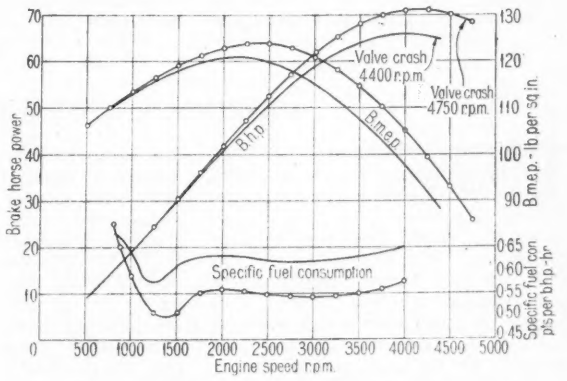


Fig. 12. Improvement in brake horse-power and thermal efficiency.

———— Three-arc cam with 4½-coil spring and 0·360-inch lift.
—○—○— Multi-sine-wave cam with 3½-coil spring and 0·390-inch lift.

with the 4½-coil spring (natural frequency, 20,400 cycles per min.), the 10th, 11th, 12th, 13th, 14th, 15th, 16th, and 17th orders give sufficient resonance to produce $\frac{3}{16}$ inch surge; in the case of the 3½-coil spring of the same gauge and mean diameter (natural frequency, 26,200 cycles per min.) used with a multi-sine-wave cam, the only harmonics exciting measurable resonance to produce $\frac{1}{64}$–$\frac{1}{32}$ inch surge, were the 13th and 14th orders; moreover, all harmonic orders below the 12th were moved out of the running range of the engine by the high-frequency spring.

In view of recently published work in America (Dudley) on the subject of cam design, it is of interest to compare the alternative solutions of the same problem. Mathematical research on the multi-sine-wave cam was commenced by the author in November 1946 following development work on the subject.

Acknowledgements

The author wishes to express his thanks to the Chairman and Directors of the Austin Motor Company, Ltd., for permission to prepare the paper, to those members of the staff who co-operated in its preparation, to his erstwhile colleague Mr. H. N. Charles, to Dr. J. H. Weaving, and to Mr. J. R. Rix for his encouragement.

APPENDIX

References

CANDEE, A. H. 1947 (October) Trans. A.S.M.E., vol. 69, pp. 709–711, "Kinematics of Disk Cam and Flat Follower."

DUDLEY, W. M. 1948 (January) Trans. S.A.E., vol. 2, No. 1, pp. 19–33, "New Methods in Valve Cam Design."

HELDT, P. M. 1944 "High-speed Combustion Engines," pp. 369–372 (P. M. Heldt, Nyack, N.Y.).

1941 edition, pp. 379–382 (Iliffe and Sons, Ltd., London).

WAHL, A. M. 1944 "Mechanical Springs," first edition, pp. 230–232 (Penton Publishing Company, Cleveland, Ohio).

BRITISH STANDARDS

B.S. 1709:1951, *Nomenclature of Drawing Instruments*, (*price* 2*s* 0*d*) has just been issued in the series relating to drawing office equipment and materials. The Standard gives names and descriptions of over eighty drawing instruments. Compasses are dealt with first, these being divided into six classes, i.e. half sets, compasses, dividers, bows, bow half sets and spring bows, each of which is defined.

The remaining instruments are subdivided into five classes, i.e. ruling pins, prickers, steel tracers, beam compasses and proportional dividers, a description of each instrument being given. The scales most commonly appearing on proportional dividers are included in an appendix.

B.S. 1723:1951, *Brazing*, (*price* 3*s* 0*d*) has been recently published by the British Standards Institution. This standard is supplementary to B.S. 1724—*Bronze Welding by Gas* (in course of preparation) and has been prepared as part of the programme of welding standards authorized by the Welding Industry Standards Committee of the Institution.

The Standard is confined to those aspects of brazing for which it is considered advisable to specify definite requirements, although general recommendations regarding the features of one or more methods of brazing are given where it is thought that guidance will be helpful. In view of the vast range of joints and connections for which brazing is used, this treatment of the subject has been considered the most appropriate.

The brazing processes covered are blow-pipe (torch), furnace, electric induction, electric resistance, dip and salt bath. A general section specifies requirements for joint design, general preparation, flux removal, inspection and testing procedure. Each process is dealt with in a separate section in terms of a description of the processes and the permissible filler metals for the various parent metals.

B.S. 1710:1951, *Colour Identification of Pipe Lines* (*price* 2*s* 6*d*). Of recent years considerable attention has been given to the use of colour for the identification of pipe lines, and a number of difficulties have arisen because of differing opinions as to whether the identification should be on the basis of the characteristic of the pipe content, or the identification of the pipe content as such.

During the past twenty years three British Standards have been published dealing with the identification of piping, and certain anomalies have arisen. In order to remove these difficulties, and to co-ordinate requirements, this code has been prepared and replaces B.S. 457, *Identification of Chemical Pipe Lines*, B.S. 617, *Identification of Pipes, Conduits, Ducts and Cables in Buildings*, and B.S. 3011, *Identification Colours for Engine Room Piping*, but it does not supersede the British Standards dealing with gas cylinders and medical gas cylinders, nor does it affect the provisions of B.S. 158, which deals with the colour identification of switchgear bus-bars and connections and wire.

The Standard itself provides primary identification colours which indicate the class of content in pipe lines, these colours being applied along the length of the pipe or in broad rings, as desired. Detailed identification of the contents by narrow rings or letters and for certain special conditions, are covered in the appendices. Consideration has shown that, since only a limited range of colours is suitable for use under all conditions without confusion being caused through fading or discolouration, any highly complex scheme for identification is unlikely to be really practicable and a simpler code has been evolved. Coloured illustrations of the application of the code are included.

Copies of these standards may be obtained from the British Standards Institution, Sales Department, 24, Victoria Street, London, S.W.1. (1959)

AMERICAN AND BRITISH PROJECTION

TOLEDO Woodhead Springs Ltd., Clifton Works, Neepsend Lane, Sheffield, 3, have issued a brochure entitled *American* (3*rd Angle*) *and British* (1*st Angle*) *Projection*, by Guy L. Murray (price 10s. 6d.). The object of this fifteen-page brochure, with its additional three-coloured illustrations on transparent plastic material, is best described by quoting the foreword by Sir Charles Inglis:

"Mr. G. L. Murray, in his brochure, draws attention to an unfortunate ambiguity which can occur in workshop drawing technique; an ambiguity which may lead to work being produced laterally reversed or even turned upside down. To avoid this confusion, American engineers have for many years past adopted a method of projection which leaves no possible doubt about a draughtsman's intention.

Mr. Murray expounds this American method with admirable lucidity and urges that in this country we should fall into line with American practice. Mr. Murray writes with authority in that his drawing office experience has been varied and extensive and he has gained a high reputation as a teacher of this most important branch of engineering science. I hope that this brochure will be widely read and that it will expedite the standardization of drawing-office practice which has been too long delayed."

Whilst it is natural to deplore any tendency to copy blindly American practices, it must be remembered that the Americans are much sought-after customers. Furthermore, in their country the customer is always right. This is a principle which seems often to have been forgotten in this land of ours. There might, therefore, be some justification for following American standard drawing office practice in view of the fact that we export many of our engineering products to them, and that the nature of such products often necessitates the dispatch of drawings with the goods.

CURRENT PATENTS

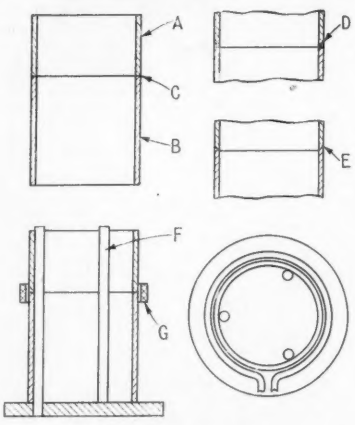
A Comprehensive Review of Recent Automobile Specifications

Two-part cylinder liner

ECONOMY in expensive alloying ingredients is secured by the use of a cylinder liner formed in two parts having different physical qualities. Only one part needs to be of a special alloy possessing resistance to abrasion or corrosion while the other may be of a conventional iron having the required receptiveness for lubricant. For convenience the two parts are united so that the liner may be handled as a single component.

Part A of the liner, required to have the greater resistance to wear, is of an iron alloy containing 1·5 per cent to 2·0 per cent carbon and 28 per cent to 34 per cent chromium. Both this and the other part B may be cast by the centrifugal process. The contiguous ends of the two parts may be flat as at C, spigoted, as at D, or formed with complementary bevelled surfaces E.

Butt welding may be used to unite the

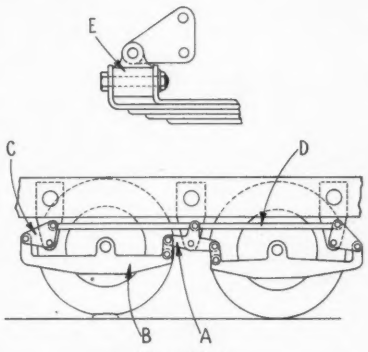


No. 640845

two parts but preferably they are joined by a high melting point brazing material such as silver solder. Conveniently they may be assembled on a jig comprising three or more rods F extending from a base to ensure axial alignment. Also supported from the jig is an electric heating coil G located around the junction of the liner parts and adapted to fuse the brazing material. *Patent No.* 640845. *British Piston Ring Co., Ltd., and T. R. Twigger.*

Suspension of twin driving axles

IN a conventional arrangement, the foremost and rearmost ends of a pair of springs on one side of the vehicle are connected to the chassis frame while the adjacent ends are linked to a balance beam pivoted on a frame bracket. As an axle can only have approximately half the vertical movement of the end of the spring linked to the balance beam, the beam is liable to be subjected to excessive angular movement for a comparatively small rise and fall of the axle. This method of coupling the spring supports circumvents that disadvantage and in operation ensures that the two axles move equal distances in opposite directions.



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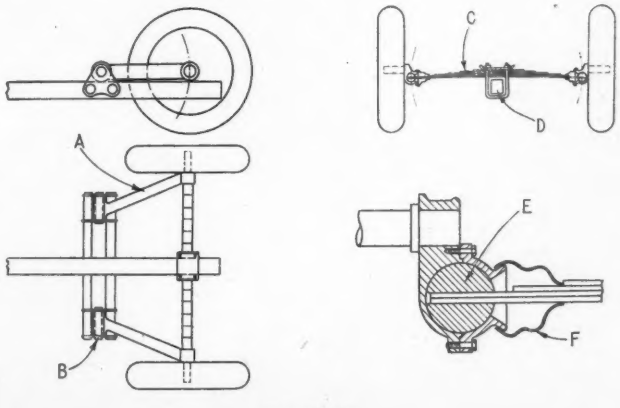
Balance beam A, pivoted on a frame bracket, is of inverted T-shape and to its lower arms are shackled the inner ends of springs B. The outer ends of the springs are shackled to oppositely arranged bell-crank levers C, also pivoting on frame brackets. Each bell-crank lever is coupled by a rod D to the vertical arm of the balance beam. Preferably the shackles are of the cross-pivot type, as illustrated at E, or may be of some alternative spherical joint arrangement to permit transverse tilting of the axles without torsionally stressing the springs.

When one wheel rides over an obstruction, bell-crank levers and balance beam all pivot in the same angular direction and the other wheel will be moved to the same extent in the opposite direction, keeping both wheels in contact with the road. *Patent No.* 640121. *J. T. Naylor and Atkinson Lorries, Ltd.*

Independent wheel suspension

IN this arrangement each wheel is carried on a crank arm A swinging about a pivot axis B perpendicular to the longitudinal axis of the vehicle. Suspension is by means of a single transversely disposed spring C clamped to the central backbone member D of the chassis. Spring ends and the ends of the crank-arm describe arcs in planes at right angles to each other and the main feature of the invention is the method of permitting this divergent movement at the connection between spring and wheel carrier.

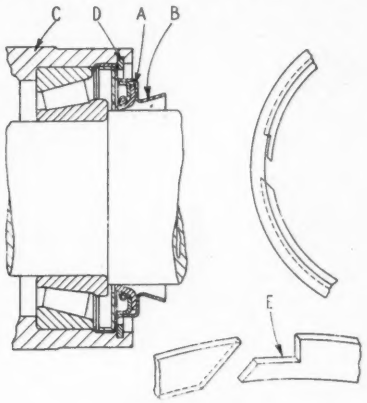
Housed in the end of the crank-arm is a slotted ball E which receives the end of the spring with freedom to slide in both longitudinal and transverse directions. As the ball can turn in any direction in its housing, no torsional or bending stresses are imposed on the spring. A flexible gaiter F is fitted between spring and ball housing to exclude dust and prevent loss of lubricant. *Patent No.* 639170. *Aviation Works and R. Vykoukal. (Czechoslovakia.)*



No. 639170

Sealing Wheel Hubs

A COMMON method of retaining the oil sealing ring of a wheel hub is by means of a lock ring but some difficulty may be experienced when it is desired to dislodge the ring in order to renew the seal. Such a mounting is shown, in which the seal A



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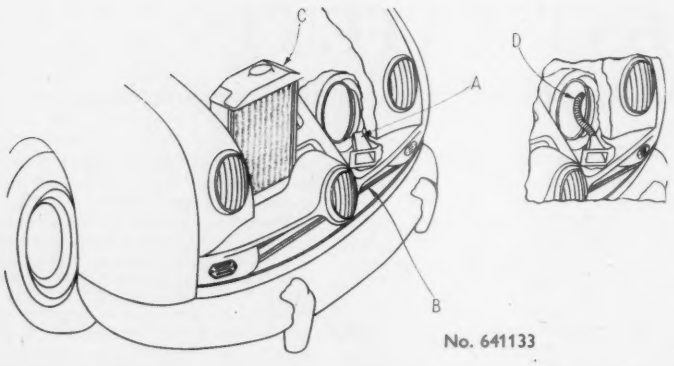
is housed in a member B secured in wheel hub C by a split retainer ring D.

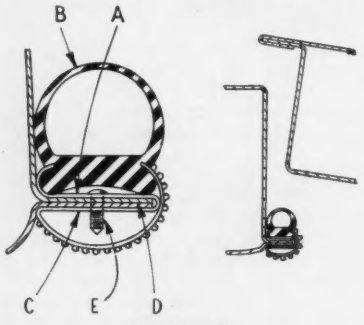
To facilitate removal, however, one end of the ring is stepped, as at E. When in position the step leaves a gap between the ring and the hub and insertion of a screwdriver enables the ring to be prised out of its groove. *Patent No.* 636076. *Ford Motor Co. Ltd. and J. H. Smith.*



No. 641133

Sealing doors and windows

IN this invention a contoured rigid metal retainer serves to hold a flexible weatherstrip around a door opening and, at the same time, present an interior trim beading. The mild steel retainer strip is generally of a narrow U-shape, one limb A constituting the base of the channel to receive the tubular resilient weatherstrip B. Forming the interior trim element, the second limb C is turned back upon itself in a smooth curve to pass beyond the U-section. The inturned margins of both limbs constitute the flanges of the channel receiving the weatherstrip. This second limb may



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be painted or covered with a cloth secured by an adhesive to produce the desired decorative effect.

To seal a complete door opening it may be necessary to fit the metal container strip in two complementary parts which will be formed to shape and size prior to installation. It may be attached to the door opening merely by forcing the U-section over the maginal body flange D but screws E or other fixing elements may be used to prevent inadvertent displacement.

In an alternative arrangement, the sheet metal retaining strip is replaced by an extruded section of light alloy. *Patent No.* 641145. *Humber Ltd., and Pressed Steel Co., Ltd.*

Cooling front wheel brakes

MODERN body construction tends to partially enshroud the front wheels and consequently to impede the flow of air passing over the brake drums and it is a known practice to provide access for a flow of air from the front of the vehicle to reach the wheels. In this invention, ducts are arranged to deliver air directly to the exterior or the interior of the brakes for cooling purposes. In one layout, a pair of air collectors A in the form of ducts mounted on the chassis or in the body are located in the space between the front grille B and the radiator C. Each is directed towards a brake so that the issuing air will impinge on the exterior of the drum or backplate.

In a modification, air collected in each of the ducts is led by a flexible pipe D through the stationary backplate in order that the brake may be cooled internally. *Patent No.* 641133. *Austin Motor Co., Ltd., J. R. Rix and J. R. Stanfield.*

Chassisless construction of heavy vehicles

RELATING to a passenger or goods vehicle, this method of construction provides for the manufacture of a self-supporting body and axle trucks as independent components. The body, indicated in broken outline, is a self-contained, transportable unit incorporating on its underside three girder-section transverse frame members A, B and C and a channel member D. The front axle truck comprises a rectangular channel section frame E, the longitudinals carrying the spring supports and the transverse elements spaced to support directly transverse members A and B on the body. For the rear axle truck a similar channel frame F is used but is of reduced longitudinal dimensions so that the spring supports are mounted in both cases under the transverse elements. Attachment to the body transverse members is directly to girder C and laterally to channel D.

In both cases the springs are arranged directly under the truck longitudinals so that those members may constitute abutments in the event of undue spring flexure. For convenience in assembly, a relatively light intermediate frame, shown at G, may be provided to connect the two axle trucks and in service may be used to carry certain vehicle auxiliary equipment.

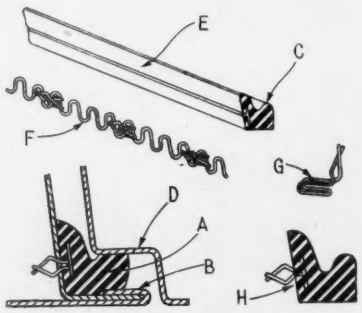
Axle trucks can be constructed in a small number of standard types which can be used for vehicles having wheel bases of different lengths. Only the unloaded, intermediate frame needs to be adapted to the required wheel base. If desired, guide plates H may be secured to one or more of the transverse body frame members to facilitate the mounting of the body on the trucks. The body is secured by two bolts at each of the transverse bearers. *Patent No.* 639100. *A. Abbey.* (*AB Scania-Vabis, Sweden.*)

Flexible weatherstrip

MODERN trends in design result in body and door contours requiring a weatherstrip which may be flexed longitudinally and transversely and, in some instances, twisted. The aim of this proposal is to provide a combined weatherstrip and retainer which can be quickly and permanently secured to a body or door surface regardless of the contour.

A resilient sealing strip A has flat surfaces adapted to be positioned in engagement with the jamb face and the marginal flange of a door B. These two surfaces are joined by a suitably profiled curve yielding a protuberant sealing flange C to engage the recessed margin D of the door opening. In surface E is formed a T-shaped groove to receive a wire insertion F of sinuous shape. Both the resilient strip and the wire insert may be formed by a continuous process and cut to any desired length.

Wire trim clips G are hooked on to the



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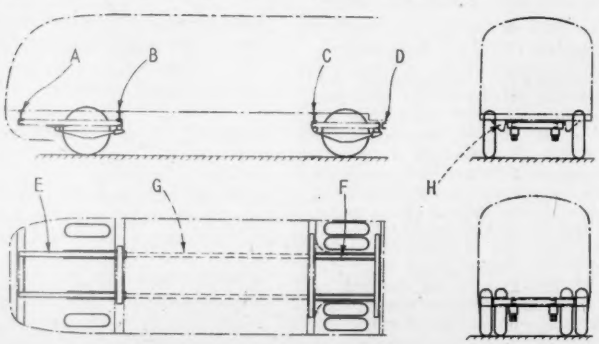
insertion at any desired spacing with the spring legs protruding through the lips of the slotted weatherstrip.

The assembled strip will flex, twist or stretch as may be required to suit the contour of the support member and is secured by snapping the legs of the trim clips through spaced holes in the jamb face of the door. As a modification, the wire reinforcement, complete with spaced trim clips, may be moulded in the resilient strip, as at H. *Patent No.* 640870 *Ford Motor Co., Ltd.*



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